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AGT-1500 Combustor and Fuel Effects Modeling

Final Technical Report

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West Lafayette, Indiana

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The influence of alternative fuels on gas turbine engine performance is considered analytically. Broad specification or alternative fuels may have viscosities and wider boiling point temperature distributions which adversely affect atomization and vaporization processes within gas turbine combustors, leading to performance penalties in ignition, lean blowoff and combustion efficiency. These performance measures have been modeled by considering

ratios of characteristic times which describe the controlling physical processes of fluid mechanic mixing, kinetics and fuel evaporation. Semi-empirical models are developed which describe lean blowoff, ignition and combustion efficiency of the AVCO-Lycoming AGT-1500 combustor and other combustors. These expressions can be used to assist design of fuel tolerant combustors, and to assess impacts associated with fuel type selection.

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EXECUTIVE SUMMARY

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I. INTRODUCTION AND SUMMARY

A. Program Objectives

The relaxation of fuel specifications and the introduction of
alternative fuels such as shale oil derivatives in order to extend liquid
fuel supplies can adversely affect gas turbine combustor performance
(Moses, 1975 and Blazowski, 1978). Thus either changes in, for example,
the combustor geometry or injector must be made to offset the fuel changes,
or penalties such as decreased combustion efficiency may have to be en-
dured. The goal of this program was to develop a modeling technique which
could be used to quantify fuel property effects on lean blowoff, ignition
and combustion efficiency in gas turbine engines, particularly the
AGT-1500; the model should also include combustor geometry and operating
conditions (pressure, airflow rate and inlet temperature). With this type
of model a combustor designer given a new fuel can compute, without re-
sorting to a costly and time consuming experimental program, whether a
combustor can accommodate the new fuel and if not, the model can suggest
changes in the combustor (such as improved injector performance) which
would be necessary.

The technique used to address this problem is the characteristic
time model. The characteristic time approach associates an appropriate
·ε, which is relatively simple to evaluate, with the important physical
processes in the combustor such as droplet evaporation, chemical kinetics
and turbulent mixing.

B. Summary of Results

In the lean blowoff work, as in ignition and combustion efficiency, the model is extended from work conducted on an experimental rig (Plee and Mellor, 1979). The model represents the blowoff limit by considering the competition between the mixing time of the flame-stabilizing shear layer and the sum of the kinetic and droplet evaporation times. Data for a Detroit Diesel Allison T-63 helicopter engine burning three fuels and several geometric variations of the AGT-1500 burning a variety of fuels are correlated with the model suggesting the model is applicable to other fuels and combustors. Of critical importance is the atomization quality of the fuel spray which is a function of fuel type and injector; these properties are contained in the model via the droplet evaporation time.

Due to the similarities between the ignition and lean blowoff phenomena, the ignition model follows directly from the lean blowoff work. The ignition form of the characteristic time model was tested with data from an experimental rig and found to properly account for fuel type, drop size, equivalence ratio, velocity and pressure effects on ignition. The model was then extended to combustor data for the T-63 with thirteen fuels and five AGT-1500 variations with four fuels. As in the lean blowoff work all the data were correlated by the model to give a "universal" ignition curve.

Again the combustion efficiency model stems from earlier work on simplified geometries (Tuttle et al., 1976; Schmidt and Mellor, 1979) and on gas turbines (Mellor 1977 a, b; Mellor and Washam, 1979). The model states that combustion inefficiency is proportional to the ratio

of a kinetic time for CO oxidation to a mixing time which represents the time available before the CO reactions are quenched. This model works quite well if the evaporation rate of the fuel is fast compared to the mixing of the fuel vapor with air. In the present report data are examined from the T-63 and AGT-1500 burning heavy (low volatility and high viscosity) fuels where the evaporation rate is slow and must be included in the model via a droplet evaporation time. Two approaches with the droplet evaporation term included in "consecutive" and "separable" characteristic time models are shown to correlate the available data equally well. The approach which is "physically" correct is not evident but investigation is continuing in that area.

II. CORRELATION OF LEAN BLOWOFF FOR GAS TURBINE COMBUSTORS USING ALTERNATIVE FUELS

A. Introduction

Plee and Mellor (1979) derived a semi-empirical characteristic time model for lean blowoff of spray flames which correlates blowoff data for three simple geometries and a variety of fuels lighter than DF-2, including propane, JP-4 and Jet A. The model represents the blowoff limit by considering the competition between kinetics and fluid mechanic mixing in the flame-stabilizing shear layer. Droplet vaporization is also an important process, and was successfully included in their model. This has been confirmed by Leonard and Mellor (1979), who examined blowoff for four additional fuels (including unleaded gasoline, JP-10 and No. 6 oil), also in a simple disc-stabilized geometry.

The present work considers extending the model to correlate the blowoff performance of gas turbine combustors. Data for a Detroit Diesel Allison T-63 combustor burning gasoline, JP-5 and Diesel fuel marine (DFM) and for several geometric variants of the AVCO-Lycoming AGT-1500 combustor burning DF-2 and several other fuels are correlated using an adaptation of the model suggested by Plee (1978). In particular, the shear-layer mixing time which characterizes blowoff is identical to the mixing time which governs CO emissions for each combustor (Mellor, 1977; Mellor and Washam, 1979; see also Section IV) and the significant influence of droplet vaporization is included. The model thus provides an alternative to traditional

methods of correlating blowoff in terms of combustor air loading (see Odgers, 1977, for example) which require calibration for different geometries and fuel types.

B. Model for Lean Blowoff

The model of Plee and Mellor (1979) is defined in terms of four characteristic times: an ignition delay time, τ_{hc}; a mixing time, $\tau_{s\ell}$, in the shear layer adjoining the flame-stabilizing recirculation zone; a fuel droplet evaporation time, τ_{eb}, determined by fuel volatility and initial droplet diameter; and τ_{fi}, a fuel injection time, related to the initial velocity of fuel drops, which determines when significant amounts of fuel may be expected to leave the shear layer as droplets. In the simplified geometries examined by Plee and Mellor (1979) and Leonard and Mellor (1979), τ_{fi} was related to the establishment of a free-stream flame zone, which enhanced stabilization. At present τ_{fi} will be assumed unimportant since in gas turbine combustors the geometry precludes establishment of such flame zones; but overly long fuel injection times may lead to droplet impingement on the combustor walls.

The ignition delay time has been defined as an inverse hydrocarbon reaction rate

$$\tau_{hc} = (10^{-4}/\phi_{pz}) \exp(21000/RT_{\phi=1}), \text{ msec.} \qquad (2\text{-}1)$$

The activation energy, pre-exponential factor and the dependence on $T_{\phi=1}$, the adiabatic stoichiometric flame temperature, are identical in the τ_{hc} definition for simple geometries (Plee and Mellor, 1979). The activation energy (cal/mole) was determined from consideration of premixed propane/air data, while the pre-exponential factor was chosen to

make τ_{hc} of order 0.1 msec. The choice of $T_{\phi=1}$ to represent temperature dependence is dictated by the presence of locally stoichiometric regions within these heterogeneous systems, which serve to limit the achievable reaction rate. The equivalence ratio, ϕ_{pz}, characterizes the overall fuel/air distribution within the primary zone. Hence it depends upon combustor geometry. This is discussed further below.

The stabilization process is viewed as occurring in the inner portion of the shear layer surrounding the combustor recirculation zone, by way of turbulent mixing of fresh reactants and hot products and partially oxidized fuel (Zukoski and Marble, 1956; Altenkirch and Mellor, 1975). The turbulent mixing process is characterized by an eddy decay time, $\tau_{s\ell}$, which is related to the macroscopic characteristic dimension and reference velocity introduced by Mellor (1977) and Mellor and Washam (1979) to model CO emissions:

$$\tau_{s\ell} = \tau_{s\ell,co} = \ell_{co}/V_{ref} \, .\tag{2-2}$$

The reference velocity is given by

$$V_{ref} = \frac{\dot{m}_a RT_{in}}{PM(\pi/4)d_{comb}^2}\tag{2-3}$$

where
$\dot{m}_a$ = air mass flow (kg/sec)

R = .08205 m^3-atm/kgmole K

T_{in} = air inlet temperature (K)

M = 29 kg/kgmole air

d_{comb} = reference diameter (m).

The characteristic length for CO emissions, ℓ_{co}, is defined by

$$\ell_{co} = \left[d_{comb}^{-1} + \ell_q^{-1} \right]^{-1} \tag{2-4}$$

where, ℓ_q, the quench length, is chosen as the axial distance from the fuel injector to the primary air addition jets (see Section IV for additional discussion).

Following Plee and Mellor (1979) for heterogeneous systems the criterion for blowoff is that the mixing time available in the inner shear layer must be sufficient to both evaporate the fuel and ignite the mixture:

$$\left(\frac{T_{\phi=1}}{T_{in}} \right) \tau_{s\ell} \sim \tau_{hc} + k\, \tau_{eb} . \tag{2-5}$$

Note that the temperature ratio is required to account for the velocity difference due to the temperature difference between the inner and outer regions of the shear layer. For convenience Eq. (2-5) is rewritten as

$$\tau_{s\ell} \sim \tau_{hc}' + k\, \tau_{eb}' \tag{2-6}$$

where the primes now denote inclusion of the temperature ratio $(T_{in}/T_{\phi=1})$ in τ_{hc}' and τ_{eb}'. The coefficient $k = 0.011$ was empirically determined by Plee and Mellor (1979) and confirmed by Leonard and Mellor (1979). It is required because each of the characteristic times is simply an order of magnitude estimate. Below it is shown this value may be retained for comparing the magnitudes of τ_{hc} and τ_{eb} for gas turbine combustors as well.

The droplet evaporation time is estimated from the "d^2-law" of Godsave (1953) based upon the initial Sauter mean diameter of the spray, d_o:

$$\tau_{eb} = d_o^2/\beta .$$

<div align="right">(2-7)</div>

The evaporation coefficient, β, accounts for fuel property influences (Kanury, 1975) and is corrected for convection (Ranz and Marshall, 1952):

$$\beta = \frac{8 \, k_g}{\rho_\ell c_{p,g}} \, \ell n(1+B)(1 + .276Re^{1/2}Sc^{1/3}) .$$

<div align="right">(2-8)</div>

Here:

k_g = gaseous thermal conductivity

$c_{p,g}$ = gaseous specific heat

ρ_ℓ = fuel density at boiling point temperature

B = transfer number

Re = droplet Reynolds number

Sc = gas phase Schmidt number.

As in the previous work a constant gas phase ambient temperature (T_{amb} = 1000 K) is assumed in order to evaluate these parameters, and Re is evaluated assuming a constant 50 m/s relative velocity between the gas and droplets. The transfer number represents the ratio of heat available from the surroundings to the heat required for vaporization, comprised of the heat of vaporization, L, at the saturation temperature, T_ℓ, and the heat required to raise the liquid to T_ℓ from its initial temperature, T_o:

$$B = \frac{c_{p,g} (T_{amb} - T_\ell)}{L + c_{p,\ell} (T_\ell - T_o)} .$$

<div align="right">(2-9)</div>

The fuel density, ρ_ℓ, is also evaluated at T_ℓ, which is further assumed to be characterized by the 50% boiling point temperature. The correlation techniques of Maxwell (1950) are used to extrapolate fuel property data at

elevated temperatures and pressures from laboratory data for each fuel obtained at near-ambient conditions (Table 2-1).

The above assumptions imply that ambient conditions are considered to be similar for many combustors, and that it is chiefly fuel properties and initial droplet size which influence vaporization times and blowoff. Evaluation of droplet sizes is crucial therefore. Unfortunately, droplet size measurements are not available. Instead, empirical estimates of droplet sizes must be used. In the present work the correlation of Hunter et al. (1974) for simplex pressure-atomizing nozzles is adapted for the AGT-1500 data:

$$d_o = \frac{90.8 \ \nu^{.3} \ (\rho_\ell Q)^{.205}}{\Delta P^{.354}} , \ \mu m \qquad (2\text{-}10)$$

where:

ν = fuel viscosity, cstk

ρ_ℓ = fuel density, g/cc = kg/ℓ

Q = fuel flow, ℓ/hr

ΔP = nozzle pressure drop, atm.

This is a significant assumption because the AGT-1500 incorporates a dual-orifice airblast nozzle. However, at the low fuel flowrates experienced near blowoff the nozzle operates on the primary orifice alone, and is assumed to behave as a simplex pressure atomizer in this mode.

Some justification for this assumption is given in Figure 2-1, which presents drop size estimates, computed by Eq. (2-10) for AGT-1500 primary nozzles (5.23 kg/hr/atm$^{1/2}$), as determined by fuel mass flow and viscosity. The main orifice is programmed to open at about 5×10^{-3} kg/sec.

Table 2-1. Fuel Property Data

Fuel	No. 4	No. 6	JP-5	DF-2	DFM	JP-5	Gas
Reference	Marchionna & Opdyke (1976)- - - -				Moses & Naegeli (1978)- -		
Carbon (wt %)	.8678	.8647	.8628	.8688	.8669	.8663	.8571
Hydrogen (wt %)	.1281	.1281	.1344	.1306	.1287	.1352	.1429
H/C Molar	1.76	1.68	1.86	1.79	1.78	1.88	2.00
Fuel/Air Stoich.	14.40	14.41	14.54	14.45	14.41	14.55	14.72
Net Heat of Comb (MJ/kg)	42.48	42.04	43.10	42.65	42.92	42.94	43.60
50% Boiling Point, (K)	573	616	481	523	547	485	380
Specific Gravity (15.6 C)	.920	.870	.815	.845	.855	.824	.733
Viscosity (cstk @ 40 C)	- - - - Note 1 - - - -				3.27	1.57	0.88
Est. Heat to Vaporize at 1 atm (cal/gm)	217	240	167	190	203	169	112

Note 1: Viscosities specified by changing fuel inlet temperature.
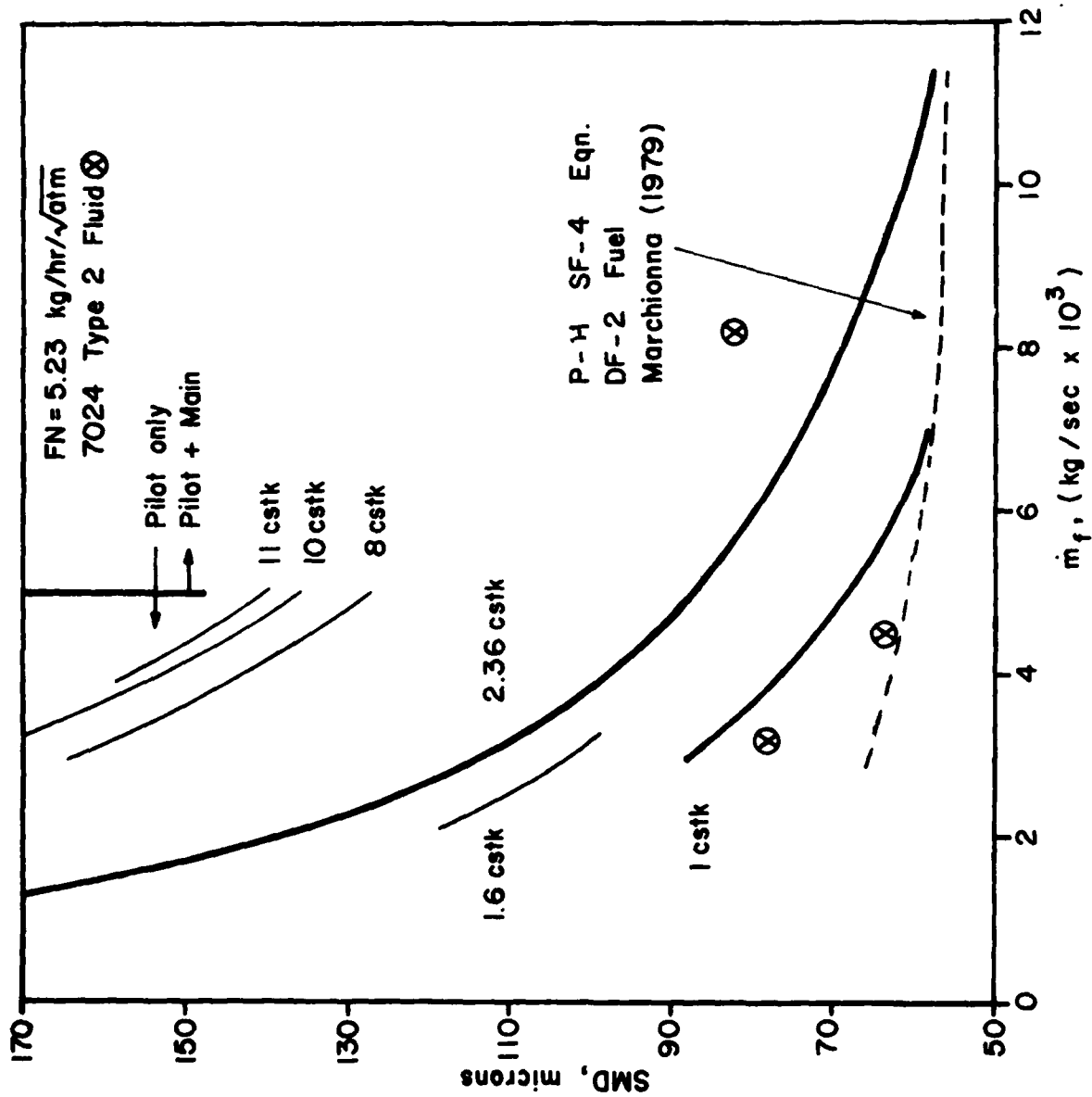
Figure 2-1. Drop size estimates for AGT-1500 nozzles and calibration data.

In this region the simplex correlation (Eq. 2-10) predicts d_o values for 2.36 centistoke viscosity somewhat larger than estimates for DF-2 obtained using a recommended correlation (dashed line) for these airblast nozzles (Marchionna, 1979). At higher flowrates the simplex and airblast correlations yield estimates in closer agreement. For pilot-only operation (fuel-flow less than 5×10^{-3} kg/sec) the simplex correlation gives estimates much larger than would be inferred from extrapolation of the airblast correlation. In this regime two calibration data obtained with 7024 Type 2 calibrating fluid are not inconsistent with predictions of the simplex correlation assuming 1 centistoke viscosity. The third calibration datum, obtained at 8.2×10^{-3} kg/sec with no airblast, indicates the substantial sensitivity to nozzle airflow. Since in the event detailed data for nozzle operation are not available, this evidence suggests the simplex correlation provides a reasonable estimate of Sauter mean diameters: the major uncertainty is sensitivity to fuel viscosity.

A different correlation cited by Moses (1975) is used to model drop sizes of the dual-orifice pressure atomizing nozzle of the T-63:

$$d_o = 140.3 \frac{\nu^{.18} (\rho_\ell Q)^{.215}}{\Delta P^{.442}} \quad (\mu m) \qquad (2\text{-}11)$$

This expression was suggested by the manufacturer; compared to Eq. (2-10) it expresses a weaker dependence on viscosity and a stronger dependence on pressure drop.

C. Experimental Data

Data from three sources have been used to check the blowoff model for gas turbine combustors. Marchionna and Opdyke (1976) examined

fuel type influences on AGT-1500 blowoff performance. Table 2-2 lists overall combustor equivalence ratios, computed from the reported fuel/air ratios at blowoff and the stoichiometry data of Table 2-1, at operating conditions corresponding to the starting and idle conditions for the AGT-1500, which is a regenerative turboshaft engine. The two combustor configurations, T-14 and T-15, differ in dilution hole size and placement, but have identical primary zone air distributions.

The unpublished data of Schmidt (1979) provide a basis for examining the influences of primary zone geometry and combustor air mass flow (Table 2-3). Four later versions of AGT-1500 combustor were tested at atmospheric pressure using DF-2 fuel. Among the six AGT-1500 variants significant changes were introduced of both primary hole location, reflected in the CO quench length scale ℓ_{co}, and primary zone air mass flow fraction, $(\dot{m}_{pz}/\dot{m}_a)$. This is illustrated in Table 2-4.

An assay of the DF-2 fuel used by Schmidt (1979) was not available. Consequently, the fuel property data for DF-2 of Marchionna and Opdyke (1976) have been adopted: on comparing this assay with others available to us, we find differences of less than 1% in stoichiometric air/fuel ratio and 4 K in computed (Svehla and McBride, 1973) adiabatic stoichiometric flame temperatures. Larger variabilities are observed in viscosity (±10%) and 50% boiling point temperature (+25/-5 K). Since these bounds generally approximate the precision of individual assay measurements, use of these representative data seems justified.

The rig test data of Moses and Naegeli (1978) were also considered. These comprised blowoff equivalence ratios obtained using Detroit Diesel Allison T-63 combustor hardware at four operating conditions

Table 2-2. AGT-1500 Blowoff Data
(Marchionna and Opdyke, 1976)

Config.	Fuel	T_{in} K	P atm	$\dot{m}_a$ kg/sec	ϕ	$T_{\phi=1}$ K
T-14	No. 4	394	1	.328	.111	2195
					.127	
					.183	
	No. 6	394	1	.328	.188	2202
					.111	
					.139	
	JP-5	533	3	.884	.033	2293
T-15	JP-5	533	3	.884	.035	2293
	DF-2				.039	2285
	No. 4				.045	2285
	No. 4				.053	2285

Table 2-3. AGT-1500 Blowoff Data
(Schmidt, 1979)

Config.	Fuel	T_{in} K	P atm	$\dot{m}_a$ kg/sec	ϕ	$T_{\phi=1}$ K
T-40	DF-2	294	1	.162	.115	2146
	DF-2	294	1	.234	.145	2146
T-50	DF-2	394	1	.331	.245	2195
		394		.398	.331	2195
		294		.167	.260	2146
T-52	DF-2	294	1	.323	.121	2146
	DF-2	294	1	.162	.176	2146
T-56	DF-2	294	1	.167	.242	2146
				.232	.225	
				.285	.222	
				.287	.234	
				.336	.274	
				.378	.433	
				.355	.361	
		394		.331	.202	2195
		394		.398	.260	2195

Table 2-4. Combustor Geometries

Combustor	ℓ_{co}, cm	$\dot{m}_{pz}/\dot{m}_a$
AVCO-Lycoming AGT-1500		
T-14	5.26	.323
T-15	5.26	.323
T-40	5.06	.293
T-50	3.21	.254
T-52	3.11	.192
T-56	3.21	.233
Detroit Diesel Allison T-63		
Std.	2.99	.25

Table 2-5. T-63 Blowoff Data
(Moses and Naegeli, 1978)

Power %	T_{in} K	P atm	$\dot{m}_a$ kg/sec	ϕ DFM	ϕ JP-5	ϕ Gas	$T_{\phi=1}$, K DFM	$T_{\phi=1}$, K JP-5	$T_{\phi=1}$, K Gas
100	547	4.77	1.10	.054	.053	.049	2314	2300	2307
55	494	3.69	.93	.069	.065	.047	2282	2268	2275
40	478	3.29	.86	.076	.067	.051	2272	2258	2265
10	422	2.30	.64	.105	.098	.063	2237	2223	2231

representative of the 10, 40, 55 and 100% power points of engine opera-
tion (Table 2-5). Although various fuel blends were considered in that
study only data for gasoline, JP-5 and DFM are examined here. The T-63
is of generally similar design to later versions of the AGT-1500 in terms
of ℓ_{co} and $\dot{m}_{pz}/\dot{m}_a$, as indicated in Table 2-4. (In fact, Section IV shows
T-63 and AGT-1500 CO emissions and combustion efficiency can be corre-
lated together.) However, the T-63 operates at nearly constant reference
velocity, a mode typical of aircraft engines. These data are included to
examine primarily fuel property and pressure influences on droplet life-
times and blowoff. Indeed, drop size estimates for each fuel obtained
using Eq. (2-11) did not vary more than ±2 microns over the four operat-
ing conditions and were taken as 85, 115 and 130 microns for gasoline
JP-5 and DFM, respectively. Estimates were also made using Eq. (2-10),
but these values seem extraordinarily high relative to previous usages of
the correlation, and are inconsistent with both our own estimates and
those of Moses (1975).

D. Results and Discussion

These data are correlated by the expression

$$\tau_{s\ell} = 2.14(\tau_{hc}' + 0.011 \ \tau_{eb}') + .22 \qquad (2-12)$$

with a correlation coefficient r = 0.91. This correlation is illustrated
in Figure 2-1. The data of Marchionna and Opdyke (1976) for No. 6 oil have
been excluded, since this fuel was observed to impinge on the combustor
walls. Also illustrated are three data, computed by Plee (1978) for T-63
operation on JP-4, Jet A, and DF-2 (Moses, 1975), which fit the correla-
tion satisfactorily.

$$\tau_{SI} = 2.13 (\tau_{hc}' + 0.011 \tau_{eb}') + .22$$

$r = 0.91$; $\sigma_y = 0.54$

35 DATA

STABLE

UNSTABLE

τ_{SI}, msec

SOURCE	ENGINE	SYMBOL
AVCO '76	AGT-1500	△
AVCO '78	AGT-1500	□
MOSES '78	T-63	○
MOSES '75	T-63	▽

Darkened Symbols Not
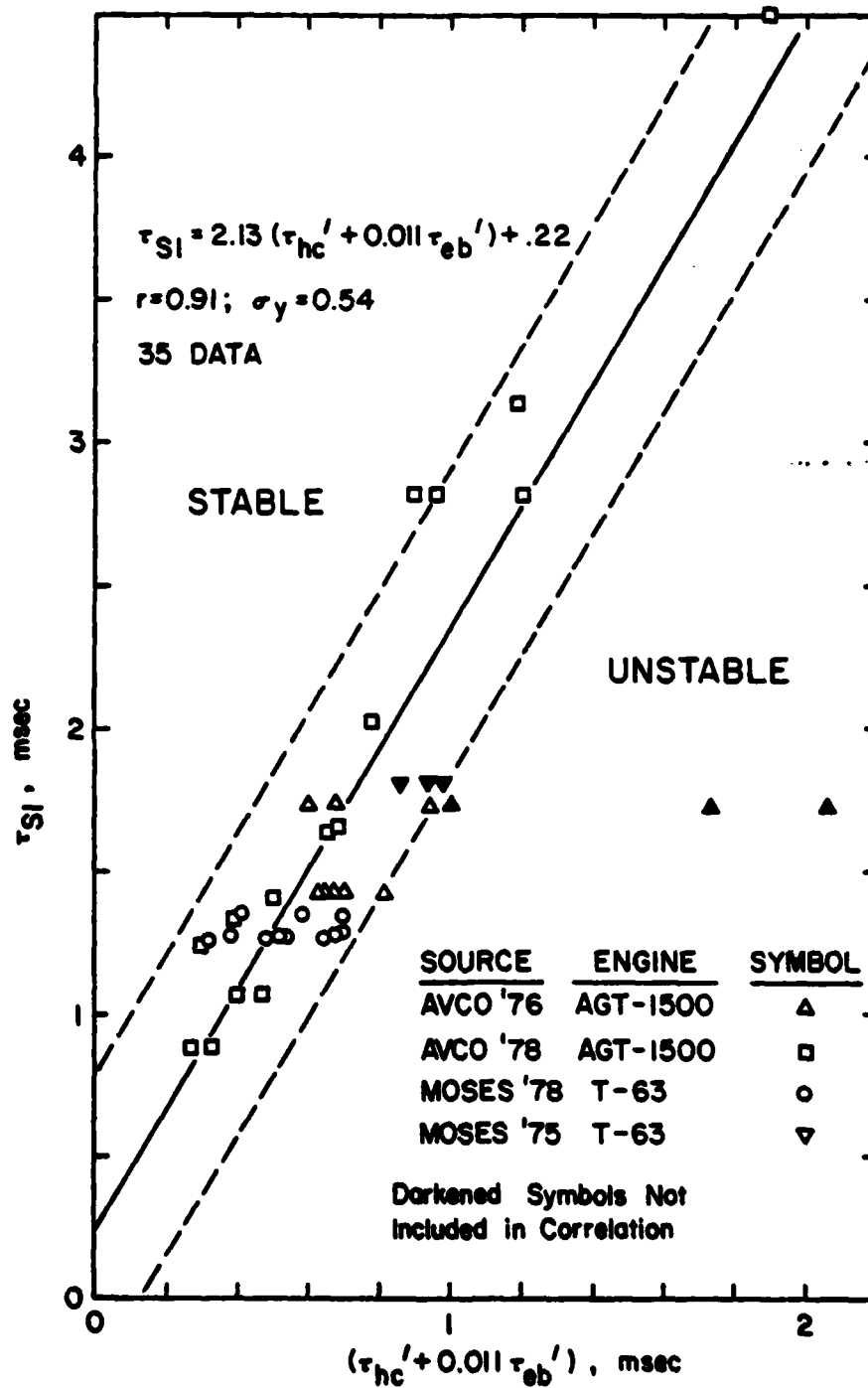Included in Correlation

$(\tau_{hc}' + 0.011 \tau_{eb}')$, msec

Figure 2-2. Model correlation of blowoff data.
Fuels include No. 4, No. 6, JP-5 and
DF-2 for the AGT-1500, and DFM, JP-5
and gas for the T-63.

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Figure 2-1 is interpreted in the following manner: increasing the available mixing time, $\tau_{s\ell}$, by reducing the reference velocity or increasing the CO quench length, will permit use of lower equivalence ratios (larger τ_{hc}) or less volatile or less readily atomized fuels (larger τ_{eb}). Alternatively, for a specified combustor and operating condition (fixed $\tau_{s\ell}$), the expected tradeoff between equivalence ratio and fuel type is uniquely specified as a required sum of τ_{hc}' and τ_{eb}'. Eq. (2-12) is not the best achievable correlation: values of the co-efficient of τ_{eb}' slightly higher than 0.011 will improve correlation quality slightly. This value has been retained, however, in order to emphasize two points. The correlation is virtually identical to that reported by Plee and Mellor (1979) to correlate bluff-body stabilized flames. This follows from choice of assumptions required to evaluate τ_{eb}' which are the same, including assumptions of ambient gas temperature, droplet relative velocity and use of Eq. (2-10) to estimate d_o. Different assumptions are expected to lead to different values of the coefficients of the correlation. If the consistency (not necessarily absolute accuracy) of drop size estimates is for the moment granted, then the existence of any acceptable correlation under the remaining assumptions argues strongly that the model of Plee and Mellor (1979) is applicable for correlation of combustor lean blowoff data. In principle the influence of combustor design modifications and fuel type selection now can be predicted by a few simple calculations. Conversely, values of the coefficients of the model are expected to be different when developed from more precise drop size estimates; but the success and utility of adaptations of the model by engine manufacturers for in-house use should

only be influenced by the consistency of drop size estimates.

Secondly, the observation that a droplet evaporation term must be included in the model mediates strongly for experimental measurement of drop sizes in conjunction with blowoff measurements. Irrespective of the adequacy of the present drop size estimates, the present work shows the failure of a simple kinetic rate model to adequately represent experimental blowoff data (since a zero coefficient of τ_{eb} will not result in an acceptable data correlation). Hence for fuel type influences ever to be understood and quantified, the spray quality must be measured and reported. And this conclusion holds irrespective of whether this model will be accepted for general use or whether more traditional correlations in terms of combustor air loading are retained.

III. SPARK IGNITION

Ignition studies by Marchionna and Opdyke (1976) conducted with the AGT-1500 and studies by Moses (1975) with the T-63 helicopter engine have shown that ignition of heavy fuels (such as No. 4 and No. 6 residual fuel oils) is difficult especially at low temperatures as noted in Fig. 3-1. Therefore the ignition process has been analyzed in an effort to develop a model which predicts the ignition limits of gas turbine combustors in order to aid combustion engineers who will be considering new fuels and combustor designs. The following sections introduce the characteristic time model (CTM) for ignition and illustrate the use of the model with data from a simplified experimental rig. Then application of the model to standard hardware is discussed in conjunction with the AGT-1500 and T-63 engines.

A. Model Derivation

The ignition model described here evolved from the lean blowoff work of Plee et al. (1978) and the previous chapter. For the ignition of a liquid fuel spray to occur, the fuel and air mixture must be heated so that the fuel evaporates, mixes with the air, and chemical reactions begin at a rate sufficient for establishing a flame; these are essentially the same phenomena which occur during flame stabilization. The difference, of course, is the energy source which initiates combustion (hot recirculating gases for flame stabilization and the spark for ignition).
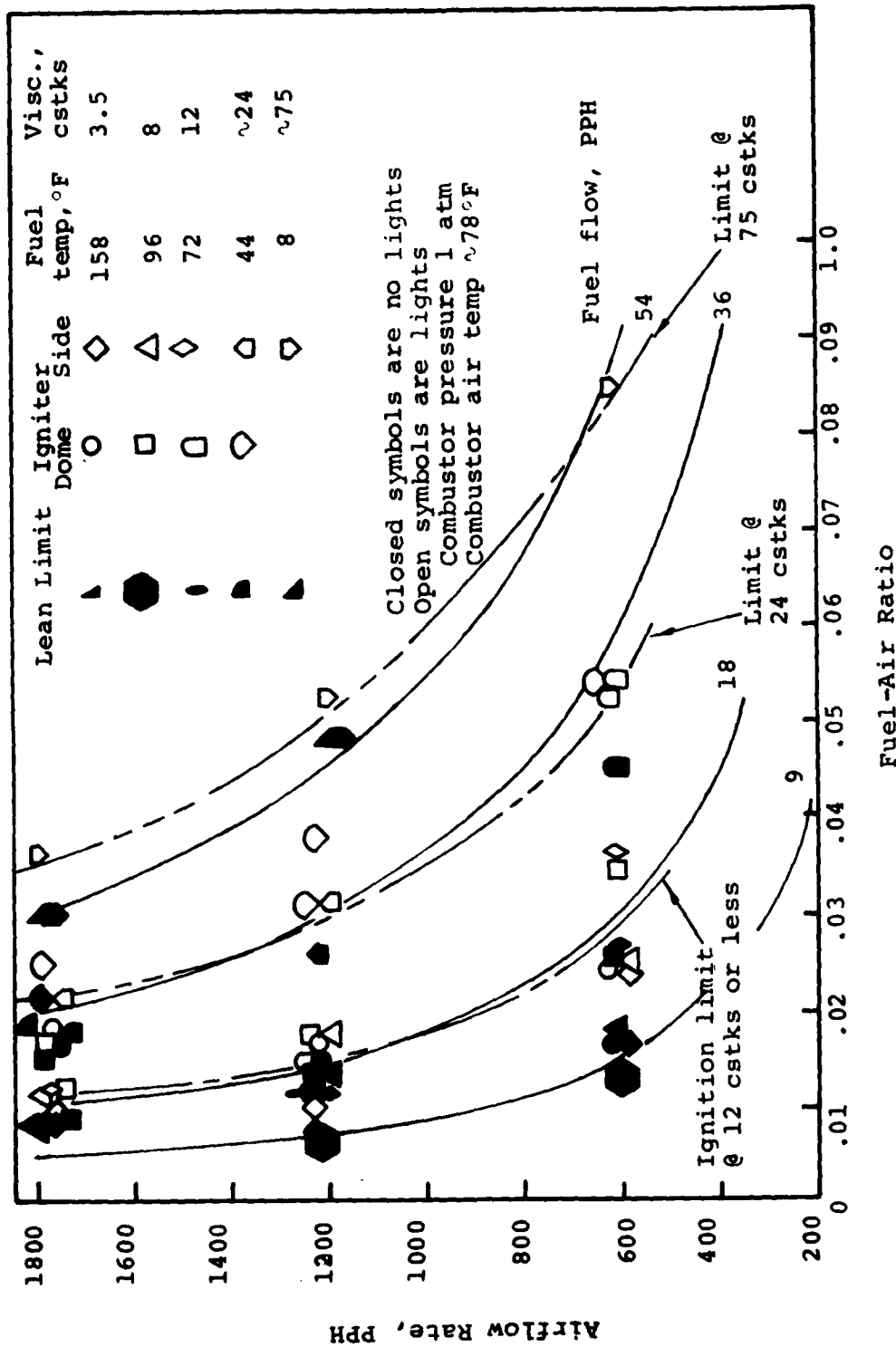
Figure 3-1. AGT-1500 ignition limit data for No. 4 fuel oil (Marchionna and Opdyke, 1976).

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Therefore, one might anticipate that an ignition model could follow from the lean blowoff work.

Initially, the lean blowoff model is assumed to apply to ignition as well. Hence, the ignition limit is established when the mixing time (now evaluated at the spark kernel) is equal to the sum of the kinetic and evaporation times. In terms of the CTM we have

$$\tau_{s\ell} \sim \tau_{hc} + a\ \tau_{eb} \qquad (3\text{-}1)$$

where, again, the proportionality and the constant weighting factor "a" are necessary since the times are simply order of magnitude estimates.

A modification to Eq. (3-1) must be made. The droplet evaporation term physically represents the time for the fuel from one "average" size drop to vaporize. Since the total amount of fuel being vaporized contributes to the ignition process and not just the fuel from one drop, τ_{eb} should be divided by the total number of drops in the spark kernel. For example if the number of drops is doubled the time required for an equivalent amount of fuel to be vaporized is halved. Noting that ϕ, the equivalence ratio, is proportional to the total number of drops in the spray we can write

$$\tau_{s\ell} \sim \tau_{hc} + a\ \tau_{eb}/\phi, \qquad (3\text{-}2)$$

which is quite similar to an ignition model presented by Ballal and Lefebvre (1980). The evaluation of each of the times for ignition is now addressed.

The length scale in the mixing time is chosen to be d_q, the diameter of the spark kernel, and the velocity is the mean flow velocity,

V, which gives

$$\tau_{s\ell} \sim d_q/V \ . \tag{3-3}$$

To evaluate d_q, a definition of minimum ignition energy is required. As suggested by Ballal and Lefebvre (1979a) minimum ignition energy is defined as the energy required to heat a spherical volume of air with diameter, d_q, to the adiabatic stoichiometric flame temperature, which gives

$$d_q = \left[\frac{E_{min}}{\pi/6 \ \rho_g \ c_{p,g} \ \Delta T_{\phi=1}} \right]^{1/3} \tag{3-4}$$

where E_{min} = minimum ignition energy,

ρ_g = density of air,

$c_{p,g}$ = specific heat of air,

and $\Delta T_{\phi=1}$ = adiabatic stoichiometric flame temperature rise.

The kinetic time is quite similar to the one used in the lean blowoff work; it is given by

$$\tau_{hc} \sim \frac{b \ \exp(E/RT_{\phi=1})}{\phi \rho_g} \tag{3-5}$$

where pressure has been included (the ρ_g term) and b is a pre-exponential factor. The activation energy for the ignition work is 26,100 cal/mole as suggested by Fenn (1951) for the ignition of propane and pentane.

To compute the droplet lifetime, Eq. (2-7) is employed. However, for ignition there are some differences in the evaluation of the evaporation coefficient. First, the convection correction used here is due to Kanury (1975) so that

$$\beta = \frac{8k_g}{\rho_\ell c_{p,g}} \ln(1+B)(0.185 \ Re^{0.6}) \qquad (3\text{-}6)$$

where Re is the Reynolds number based on drop diameter and evaluated with the reference velocity. This change was made because it resulted in an improved correlation of Ballal and Lefebvre's (1979b) data (to be examined momentarily). Another difference involves the evaluation of the boiling point temperature of the fuel. In the lean blowoff work the 50% boiling point temperature was used; however, based on the correlation of ignition limits with the 10% point by Foster and Straight (1953) and Lefebvre et al. (1978), the 10% temperature is used here. Finally, all gas properties are assumed to be those of air evaluated at an average temperature of 1300 K.

B. Experimental Rig Data

We are now in a position to test the model developed in the previous section with data from Ballal and Lefebvre (1979b) taken on a simplified experimental rig.

The data from their experiments were used to obtain d_q and the characteristic times are calculated from Eqs. (2-7), (3-3) and (3-5); the results are shown in Fig. 3-2. Note that the linear fit of the model is quite good (correlation coefficient of 0.98 and a small standard deviation as indicated by the dashed lines) over a wide range of conditions which include pressures from 0.2 to 1.0 atm, drop sizes from 20 to 170 μm, equivalence ratios from 0.43 to 1.0 and approach air velocities from 15 to 40 m/s.

Included in the figure is an illustration of how changes in starting conditions can move one from the region of ignition to a position in the "no ignition" area. For example, a decrease in equivalence ratio
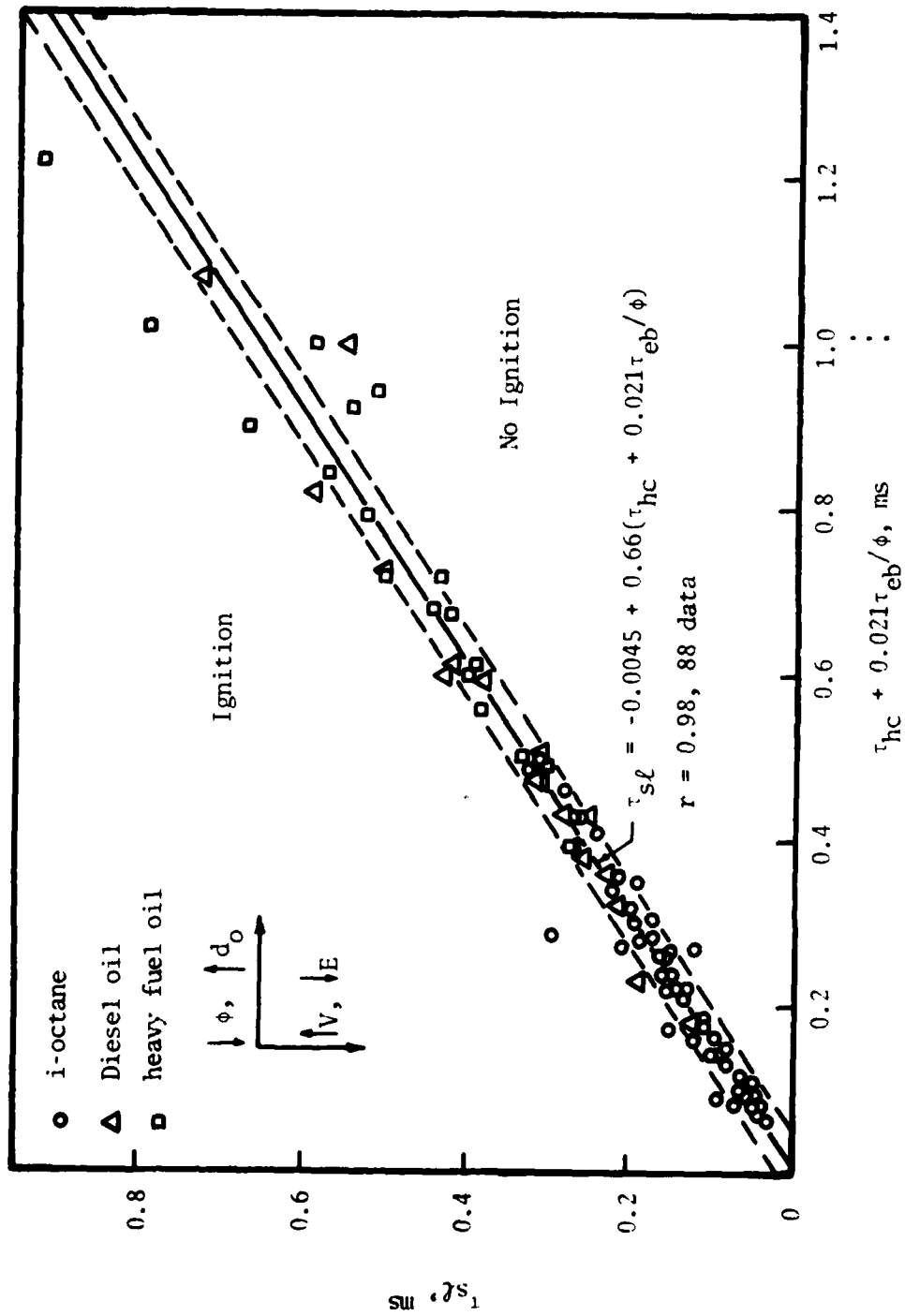
Figure 3-2. CTM correlation of ignition data from Ballal and Lefebvre, 1979b.

$$\tau_{s\ell} = -0.0045 + 0.66(\tau_{hc} + 0.021\tau_{eb}/\phi)$$

$$r = 0.98, \ 88 \ \text{data}$$

($\downarrow\phi$) or increase in drop size ($\uparrow d_o$) causes one to horizontally approach the ignition limit line. Obviously, this figure and its application are analogous to the lean blowoff curve of Fig. 2-2.

To observe the effect of drop size, velocity or any other parameter on ignition (and how well the model accounts for them) Figs. 3-3 to 3-7 are included. These figures are plots of minimum ignition energy versus a variety of parameters; the data are from Fig. 3-2 and the curves were obtained by expanding the terms in the best fit line,

$$\tau_{s\ell} = -0.0045 + 0.66(\tau_{hc} + 0.021\ \tau_{eb}/\phi)\ , \qquad (3\text{-}7)$$

which gives

$$E_{min} = \frac{0.15\ V^3\ \rho_g\ c_{p,g}\ \Delta T_{\phi=1}}{\phi^3}\ .$$

$$\cdot \left[\frac{10^{-5}\ \exp(E/RT_{\phi=1})}{\rho_g} + \frac{0.014\ \rho_\ell c_{p,g}\ d_o^2}{k_g\ \ell n(1+B)Re^{0.6}} \right]^3 \qquad (3\text{-}8)$$

where the y-intercept has been dropped.

In Fig. 3-3 the strong influence of drop size on ignition energy as indicated by Eq. (3-8) is shown. This point needs to be emphasized for two reasons. First, any changes in nozzle design or fuel properties (such as an increase in viscosity which may accompany alternative fuels) that increase the drop size may lead to ignition problems because of the very strong effect shown in Fig. 3-3. Also since drop sizes in gas turbines are usually estimated by empirical equations as discussed in the preceding section, any application of an ignition model to actual engine data may be hampered by errors in drop size estimates which can lead to large errors

Figure 3-3. Minimum ignition energy versus drop size (data from Ballal and Lefebvre, 1979b).

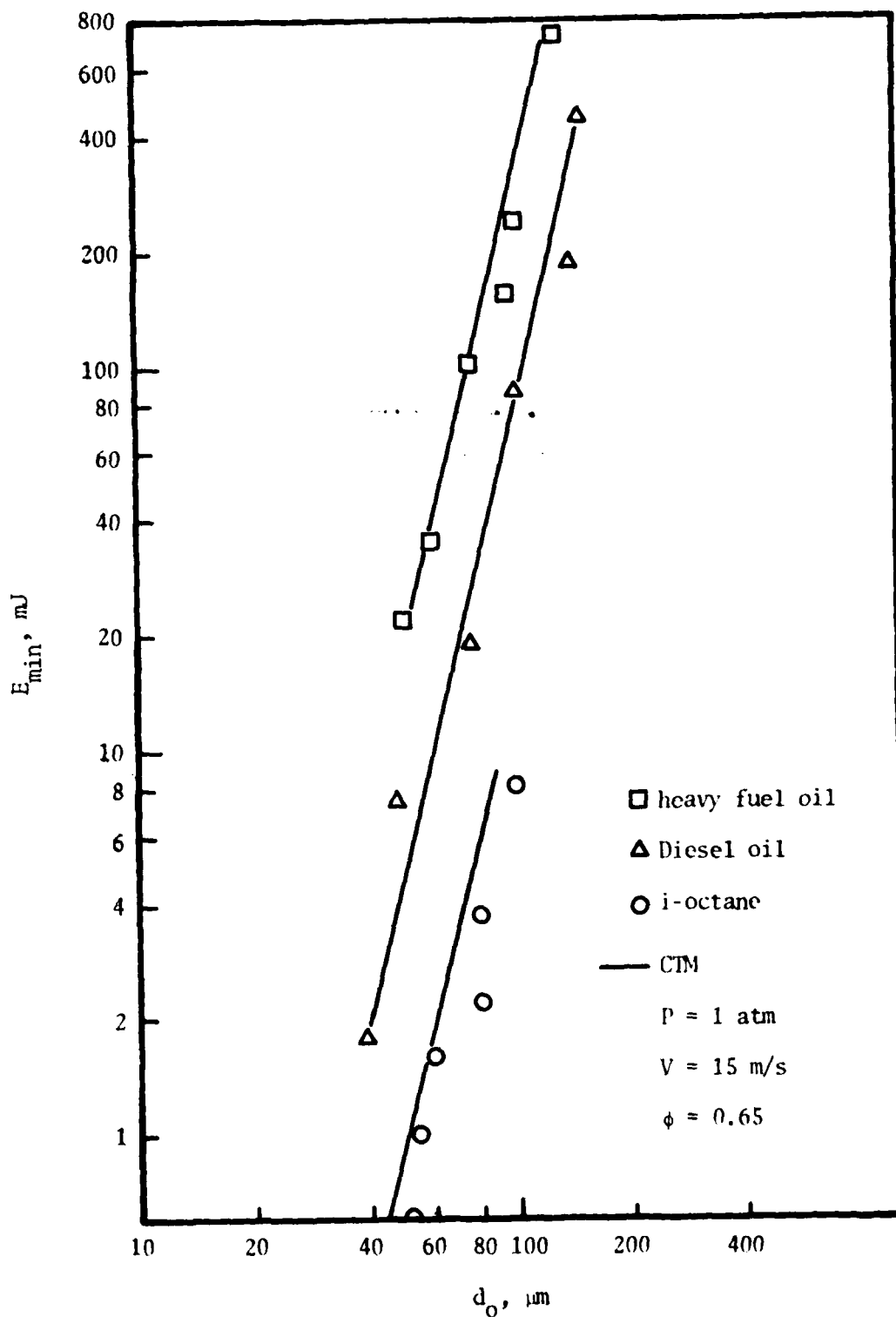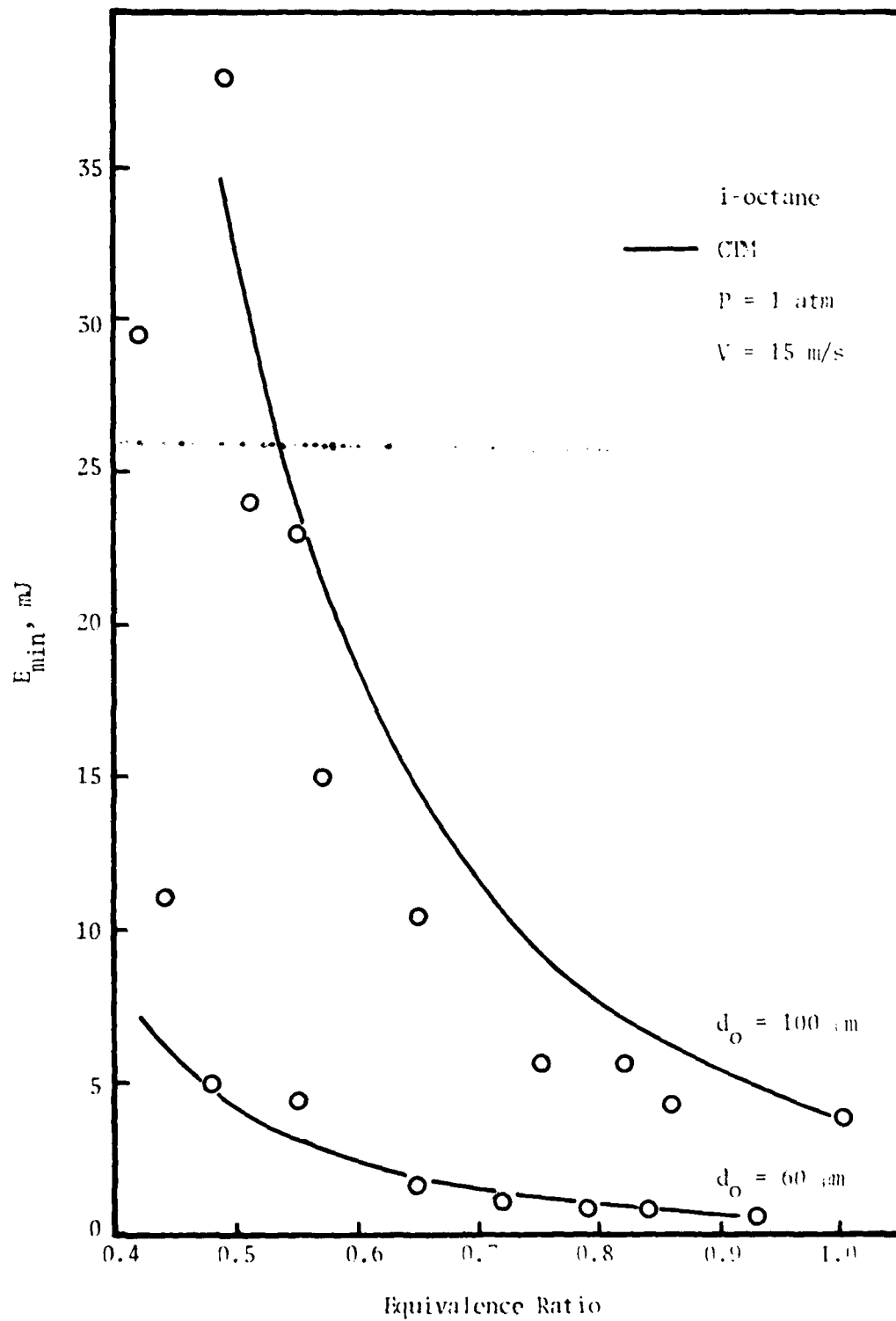Figure 3-4. Effect of equivalence ratio on minimum ignition energy (data from Ballal and Lefebvre, 1979b).
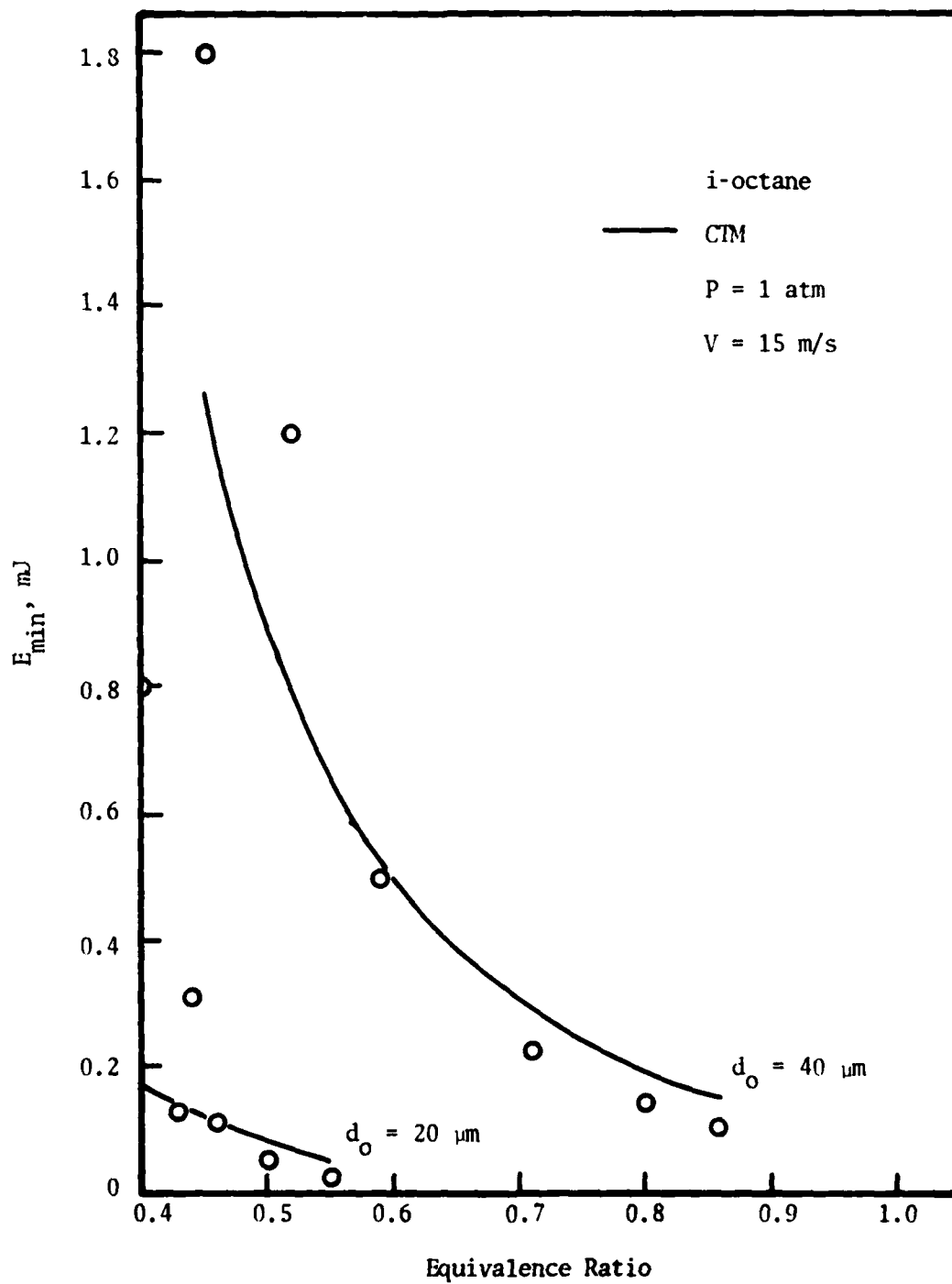
Figure 3-5. Effect of equivalence ratio on minimum ignition energy (data from Ballal and Lefebvre, 1979b).
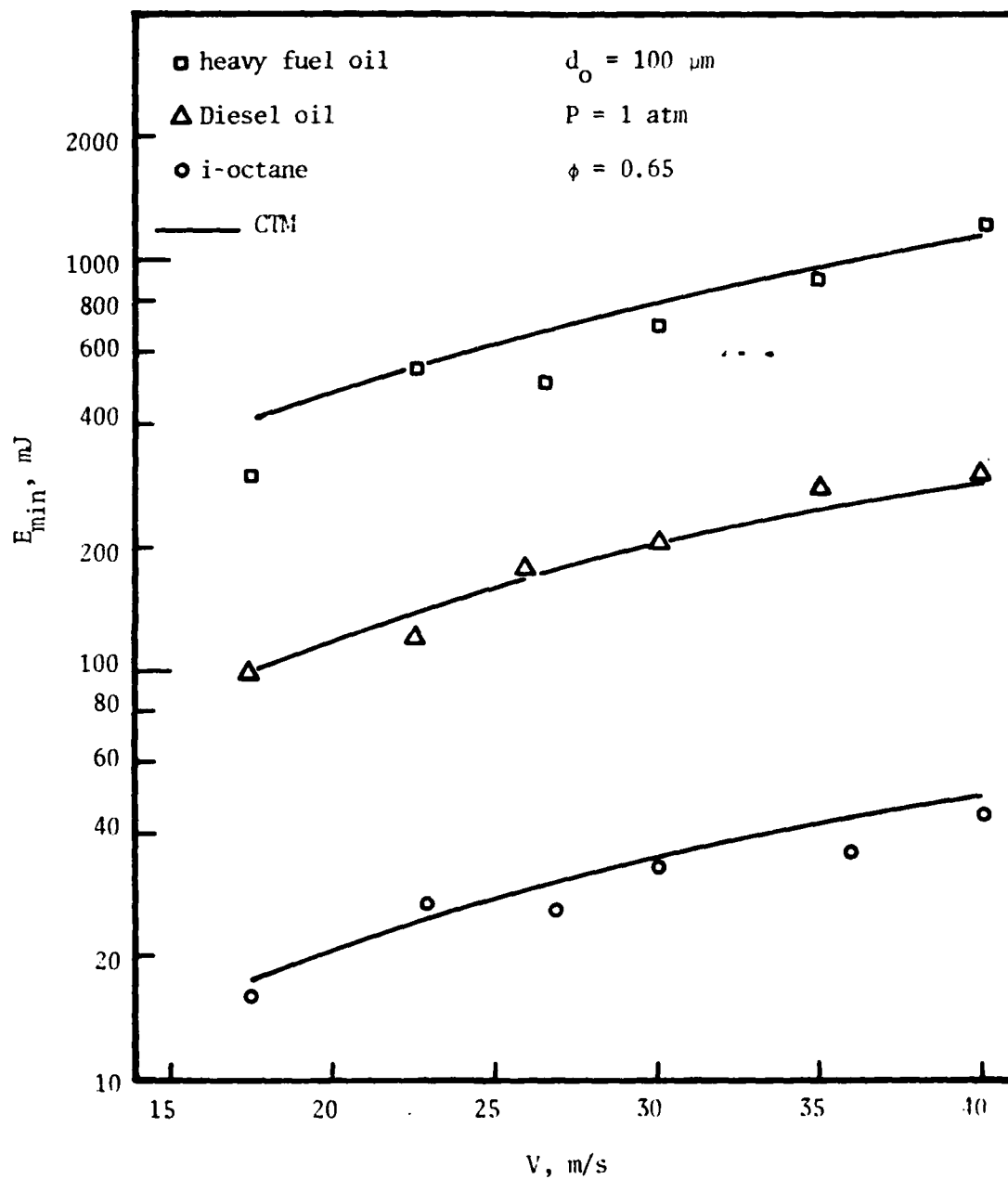
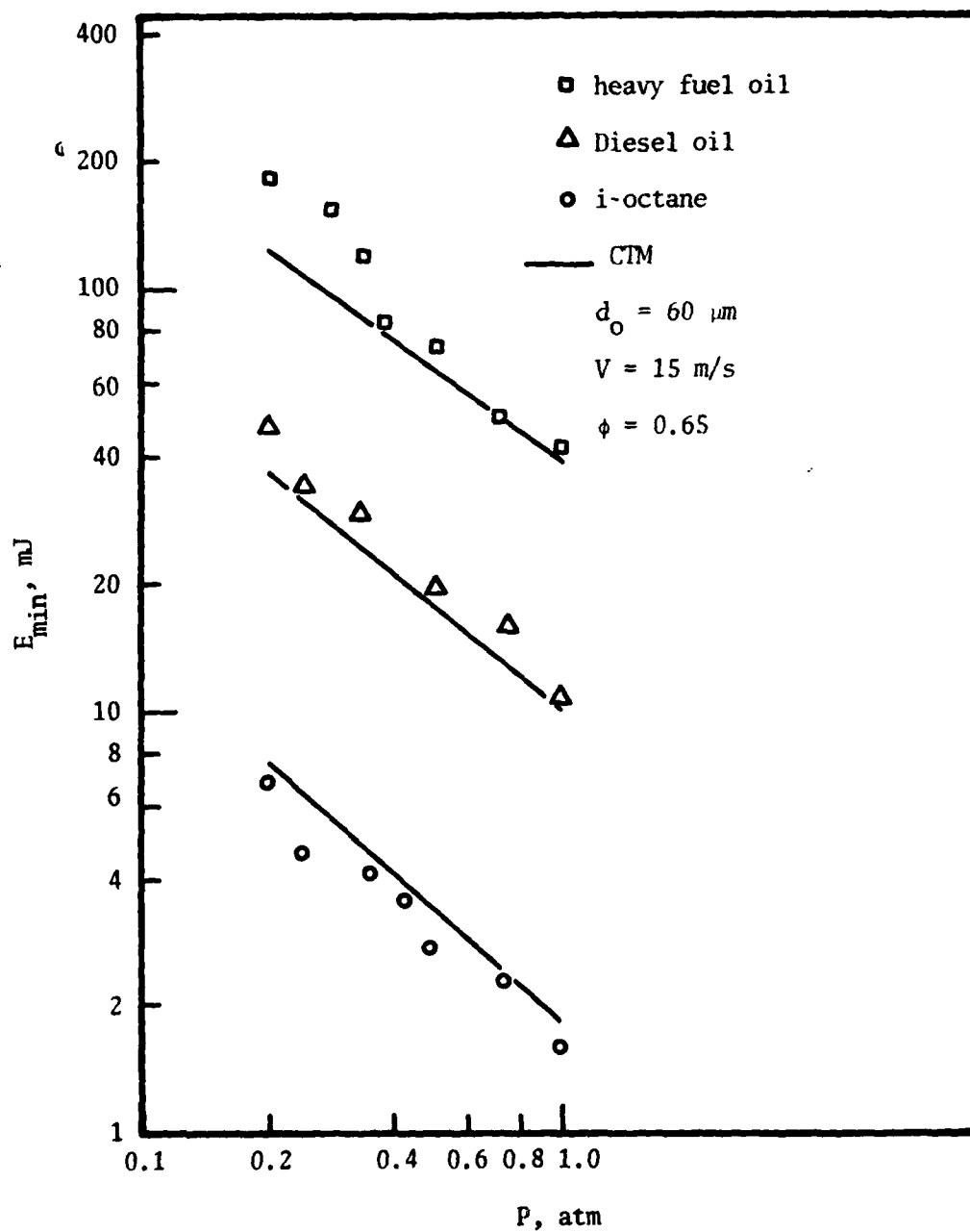Figure 3-6. Influence of velocity on minimum ignition energy (data from Ballal and Lefebvre, 1979b).

Figure 3-7. Effect of pressure on minimum ignition energy (data from Ballal and Lefebvre, 1979b)

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in ignition limit calculations.

Figures 3-4 and 3-5 illustrate the effect of equivalence ratio on ignition. Again the model agrees with the data; therefore the addition of equivalence ratio in the droplet evaporation term in Eq. (3-2) is substantiated. (Including the equivalence ratio in this manner for lean blowoff is presently being considered.)

Finally, Figs. 3-6 and 3-7 show that the model properly accounts for the effects of velocity and pressure on ignition, respectively. As the model indicates minimum ignition energy increases with increasing velocity. The influence of pressure is complex because the boiling temperature of the fuel, the flame temperature and gas density are functions of pressure. However, the net effect of increasing minimum ignition energy with decreasing pressure is correctly correlated by the model.

Thus, in this section we have seen that the CTM for ignition of liquid fuel sprays correlates the ignition data from a simplified rig quite well and that the drop size of the fuel spray is a critical factor. Now we turn our attention to the application of the model to the AGT-1500 and the T-63.

C. Application to Conventional Combustors

To apply the ignition model to combustors we must reconsider how to calculate each of the terms in Eq. (3-2). The mixing time is computed with Eq. (3-3). However, since the velocity in the spark gap is typically not known it is assumed proportional to the reference velocity and for computational purposes

$$\tau_{s\ell} = d_q/V_{ref} . \tag{3-9}$$

The quenching distance is evaluated from Eq. (3-4) with the rated energy of the spark plug substituted for E_{min}.

No changes are required in the calculation of droplet lifetime; the discussion in section A of this chapter also applies to conventional combustors. The difficulty in determing τ_{eb}, as mentioned earlier, is due to the uncertainty of the drop sizes of the fuel sprays. Consequently, for the combustors examined here the equations used to estimate drop size will be discussed for each combustor.

Finally, the calculation of τ_{hc} is given by Eq. (3-5). However, we must decide on how to estimate the equivalence ratio in the spark gap (which is the equivalence ratio in τ_{hc} and the one used to divide τ_{eb}). One logical solution is to assume that the equivalence ratio in the spark gap is proportional to the primary zone equivalence ratio and then to use the primary zone equivalence ratio for the computations. However, this approach was tried and did not work. The results of this attempt indicated that the equivalence ratio in the spark gap did not vary as the primary zone equivalence ratio but rather the spark gap equivalence ratio appeared to be nearly constant. In other words, when the primary zone equivalence ratio was used, for a given startup condition (constant $\tau_{s\ell}$) a wide range of values for the parameter τ_{hc} + a τ_{eb}/ϕ was found rather than a constant value as the model dictates. This may indicate that as more fuel is added to the combustor to enhance ignition the additional fuel does not reach the spark gap; most of it is probably swept downstream while the remainder is caught in the recirculation zone. Ignition is still improved with the addition of more fuel, however, since as fuel flow increases drop size decreases. Therefore, a constant value for the spark gap equivalence ratio was chosen. For lack of information on the amount

of fuel in the spark gap an equivalence ratio of unity was used. The assumption of a constant equivalence ratio and the selection of unity will be examined in further detail presently.

In summary, the combustor design is incorporated via the mixing time which contains the reference velocity and spark energy; nozzle performance and fuel properties are contained in the droplet lifetime. Now the AGT-1500 and T-63 are considered.

Ignition data from Marchionna and Opdyke (1976) and Schmidt (1979) include four different fuels, JP-5, No. 4 and No. 6 residual fuel oils and DF-2; two igniter locations, dome and combustor wall; and four combustor geometries (listed in Table 2-4), T-40, T-52, T-56 and the T-13 (similar to T-14). In addition, Marchionna and Opdyke (1976) varied fuel temperatures to produce viscosity changes in their fuels.

Drop size estimates for these data were obtained from Jasuja (1978) with

$$d_o = \frac{89.6 \; \sigma_f^{0.6} \nu_f^{0.16} \dot{W}_f^{0.22}}{\Delta P_f^{0.43}} \tag{3-10}$$

where
d_o = drop diameter in m,

σ_f = fuel surface tension in dyne/cm^2,

ν_f = fuel viscosity in cs,

$\dot{W}_f$ = fuel flow rate in kg/hr,

and
ΔP_f = fuel pressure drop across the injector in atm.
This equation was chosen because the drop sizes of Ballal and Lefebvre's (1979b) data were obtained with an optical technique based on the forward scattering of light due to spherical particles; the same technique was used by Jasuja (1978) to derive the empirical equation given above.

Therefore the drop sizes in the AGT-1500 are estimated with an equation derived from the same measurement technique that was used in the data which provided the initial verification of the CTM for ignition. Also note that Eq. (3-10) is for a pressure atomizing nozzle and as discussed in the lean blowoff work at the low fuel flowrates for lean blowoff and ignition the AGT-1500 injector operates in the pilot (pressure atomizing) mode only.

Figure 3-8 is the CTM correlation the AGT-1500 data and as the model suggests the data do converge to a linear fit. The correlation is not as good as the correlation from Ballal and Lefebvre's (1979b) data although it is still acceptable ($r > 0.9$). The scatter of the data about the correlation is probably due mostly to errors in drop size estimates since the drop size is such a critical parameter.

Now let us consider our assumption of a constant equivalence ratio. The data in Fig. 3-8 contain primary zone equivalence ratio variations from 0.39 to 5.3 and it is unlikely that the equivalence ratio in the spark gap is exactly constant. However based on the ignition data the change in equivalence ratio in the spark gap is not believed to be nearly as large as the primary zone change plus small differences of equivalence ratio in the spark gap are overshadowed by drop size variations. Also note that changes in the amount of air added to the primary zone in the different combustor configurations are also correlated by the constant equivalence ratio approximation. Therefore, the assumption that the equivalence ratio is nearly constant is acceptable. Of course, radical changes to igniter location such as extending the igniter directly into the fuel spray would not be correlated by the constant equivalence ratio
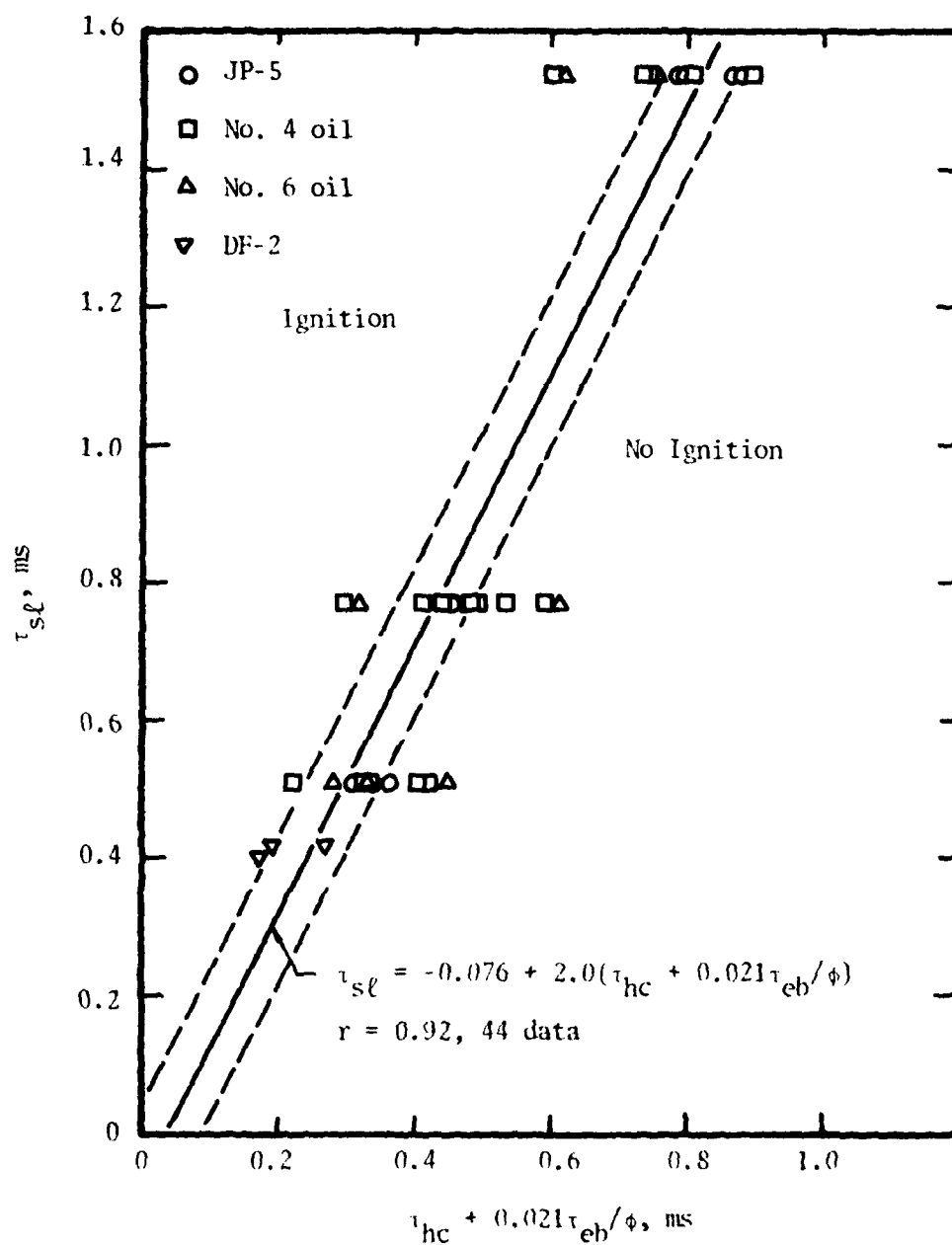
Figure 3-8. CTM correlation of AGT-1500 ignition data (DF-2
data from Schmidt, 1979; remaining data from
Marchionna and Opdyke, 1976)

assumption.

Is the correlation of Fig. 3-8 applicable to other combustors
or is it limited to the AGT-1500? This question is answered in part by
examining data from Moses (1975) and Moses and Naegeli (1978). Their
ignition tests were conducted on the T-63 helicopter engine (can-type
combustor with the igniter located in the dome) with 13 different fuels
ranging from gasoline to Diesel fuel at two different air flow rates.
The CTM computations for the T-63 data followed the procedure developed
for the AGT-1500. The one exception was that the drop sizes were esti-
mated using the empirical equation for a dual-orifice nozzle discussed
in the lean blowoff chapter. Jasuja's (1978) equation was not used be-
cause it was developed from a single-orifice pressure atomizing nozzle
rather than a dual-orifice nozzle.

The T-63 data are plotted in their CTM form, with the AGT-1500
data, in Fig. 3-9. The most important feature of this curve is that the
T-63 and AGT-1500 data can be correlated by a single equation; this
is an indication that the correlation may be applied to other standard
combustors. The scatter of the T-63 data about the correlation is similar
to that of the AGT-1500 with the exception of the two data on the left at
a $\tau_{s\ell}$ value of 1.17 msec; they correspond to gasoline and a blend of 30%
gasoline with Diesel fuel. The fuel flowrates and viscosities for these
two data are very low and it is possible that the data fall off the curve
because the empirical equation for drop size, Eq. (2-11), may be in error
at such low fuel flow and viscosity conditions. (Again recall from Fig.
3-3 that a small change in drop size can influence ignition significantly.)

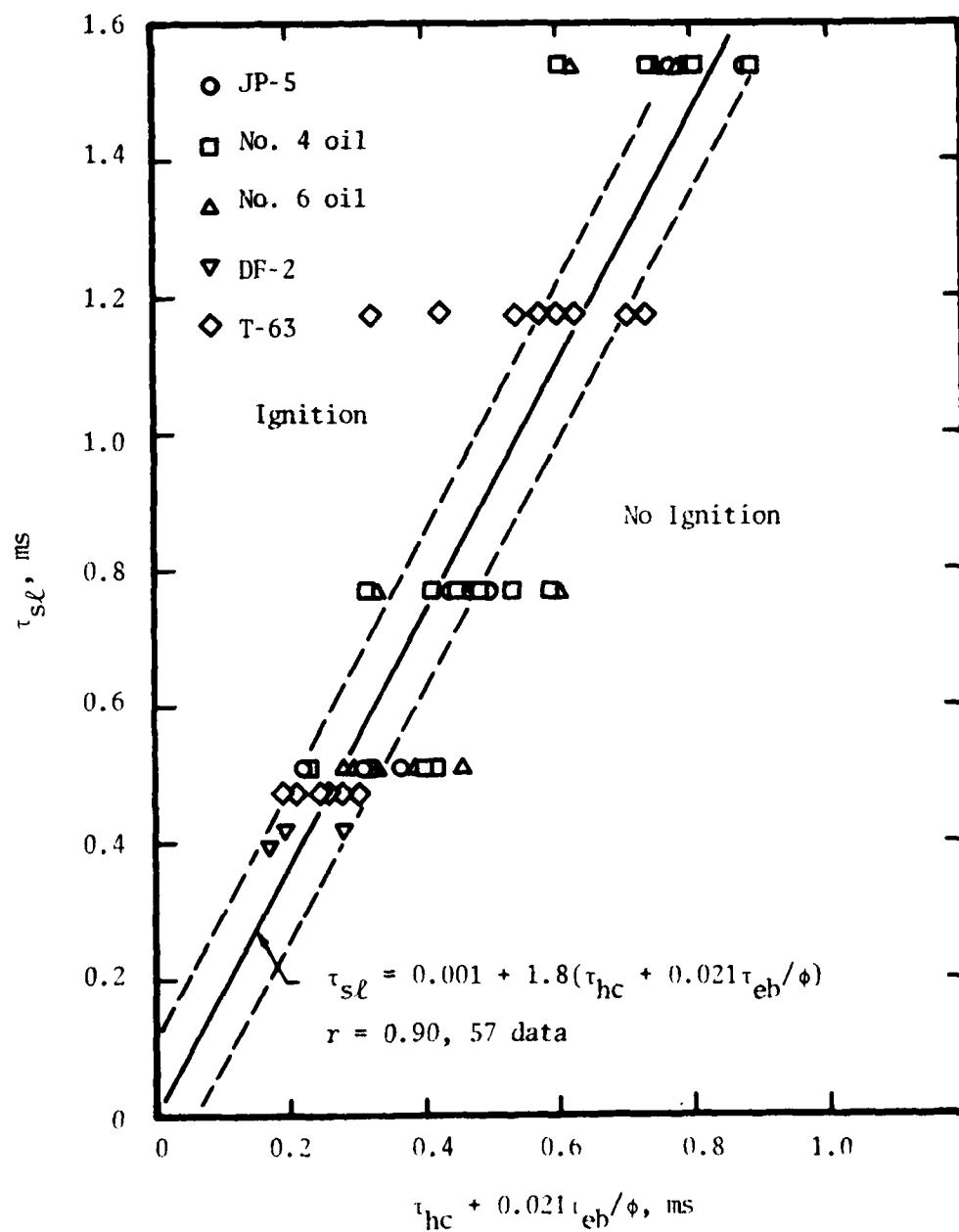Finally, notice that the slope of the correlation in Fig. 3-9

Figure 3-9. CTM correlation of AGT-1500 and T-63 ignition data
(T-63 data from Moses, 1975 and Moses and Naegeli,
1978; DF-2 data from Schmidt 1979; remaining data
from Marchionna and Opdyke, 1976)

for the AGT-1500 and T-63 is not equal to the slope in Fig. 3-2 for Ballal and Lefebvre's (1979b) data. This is due primarily to the fact that since the equivalence ratio in the AGT-1500 and T-63 was considered constant, a value of unity was arbitrarily chosen. Also the reference velocity was used as the velocity at the spark gap for the combustor data; however, in reality the velocity at the spark gap is probably only proportional (not equal) to the reference velocity. These factors result in a different slope for the combustor correlation but this difference can easily be corrected by selecting a constant equivalence ratio of 0.36 instead of unity for the AGT-1500 and T-63. The resulting correlation, including Ballal and Lefebvre's (1979b) data is shown in Fig. 3-10. This is not necessarily an improvement in the correlation of the AGT-1500 and T-63 but it allows one to produce a "universal" curve which includes data from a well defined experimental rig and actual combustor hardware.

In conclusion the characteristic time model for lean blowoff was extended to ignition. The ignition model was tested with data from an experimental rig and then used to correlate ignition data from the T-63 and AGT-1500. All the combustor and experimental data were found to fit a single curve suggesting that this may be of a "universal" nature of use to combustor designers and engineers concerned with the development of new combustors and the ignitability of new fuels in existing hardware.
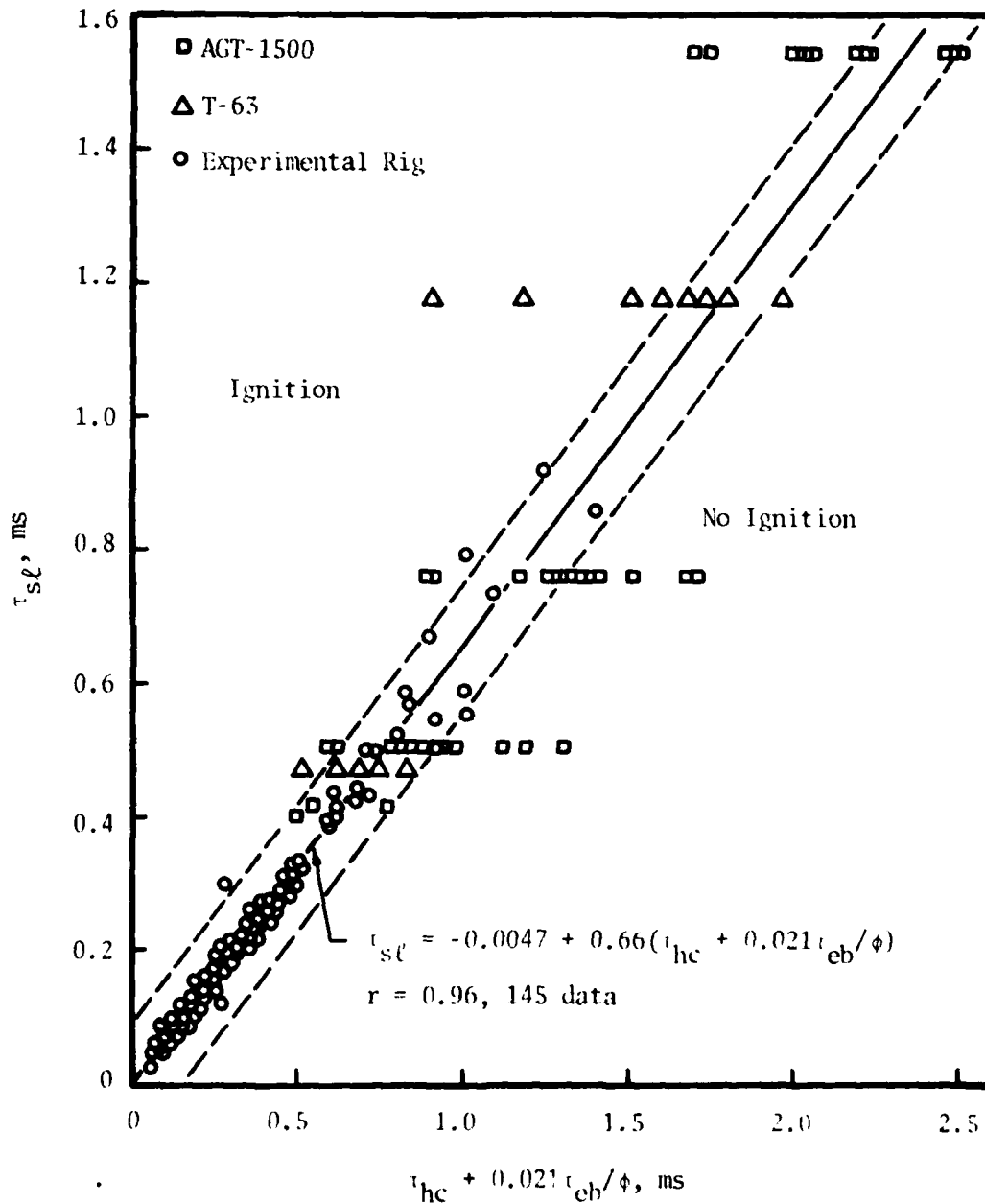
Figure 3-10. Combined CTM ignition correlation (AGT-1500 data from Schmidt, 1979 and Marchionna and Opdyke, 1976; T-63 data from Moses, 1975 and Moses and Naegeli, 1978; experimental rig data from Ballal and Lefebvre, 1979b)

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IV. COMBUSTION EFFICIENCY

A. Introduction

Fuel type selection also influences combustion efficiency. For the AGT-1500 operating at conditions above idle combustion efficiency exceeds 99.5%, and fuels as heavy as DF-2 do not significantly degrade efficiency, as shown in Fig. 4-1 (Marchionna, 1978). For smaller engines, especially those operating on non-regenerative cycles, combustion efficiencies are lower, and the influence of fuel type is more pronounced. This is illustrated by the T-63 rig test data of Moses (1975) shown in Fig. 4-2. For this combustor, efficiencies exceeding 99% are achieved only at 100% power; at low power efficiency is observed to decrease and to depend strongly on fuel type. This trend is of significance especially for small gas turbines (i.e., auxiliary power units) which may be required to accept alternative fuels in field use.

The present effort has combined elements of previous emissions correlations for gas turbine combustors (Mellor, 1977a, b; Mellor and Washam, 1979) and combustion efficiency correlations in combustors of simplified geometry (Schmidt and Mellor, 1979) to achieve an efficiency correlation for conditions when mixing-controlled quenching of kinetic processes dominate inefficiency. Under these conditions, of which the data of Fig. 4-1 are representative, heterogeneous processes are not limiting to combustion efficiency and fuel type selection shows little influence. The efficiency correlation of the AGT-1500 data was also
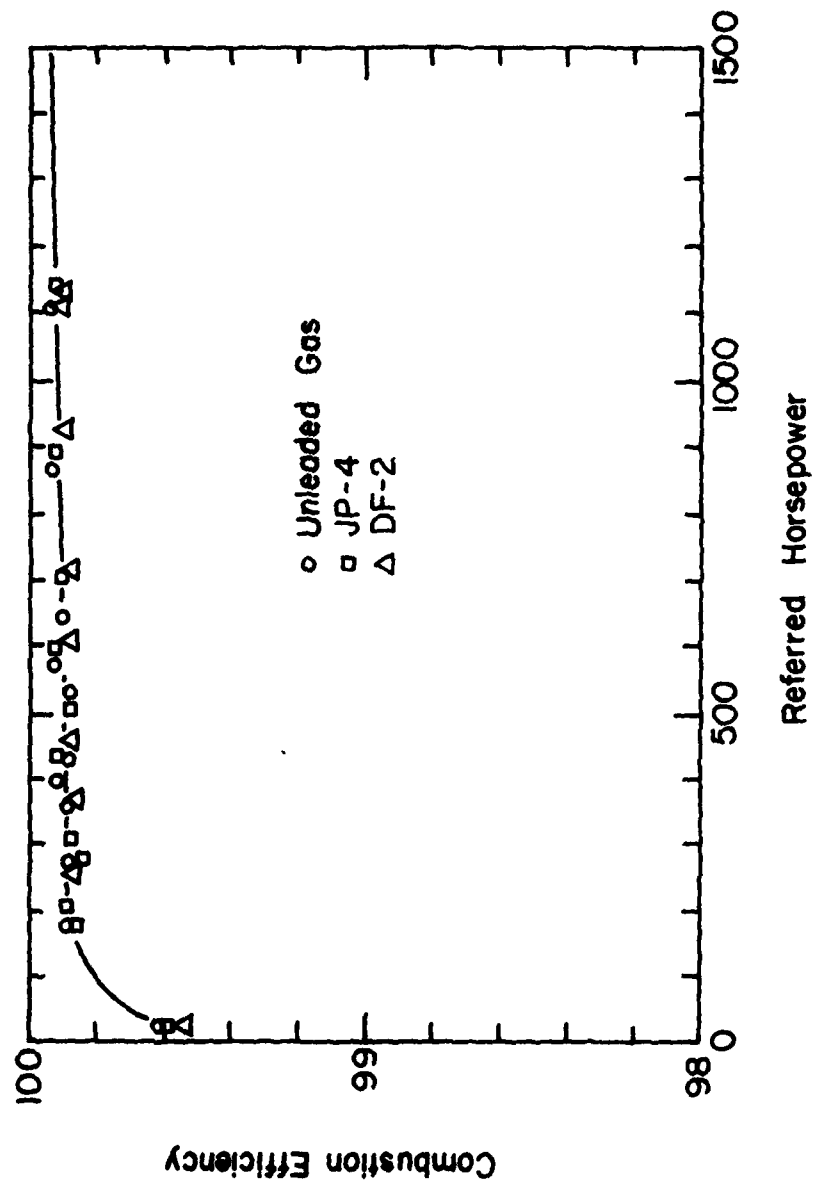
Figure 4-1. Multifuel combustion efficiency for the AGT-1500 (T-36) (Marchionna et al., 1978).
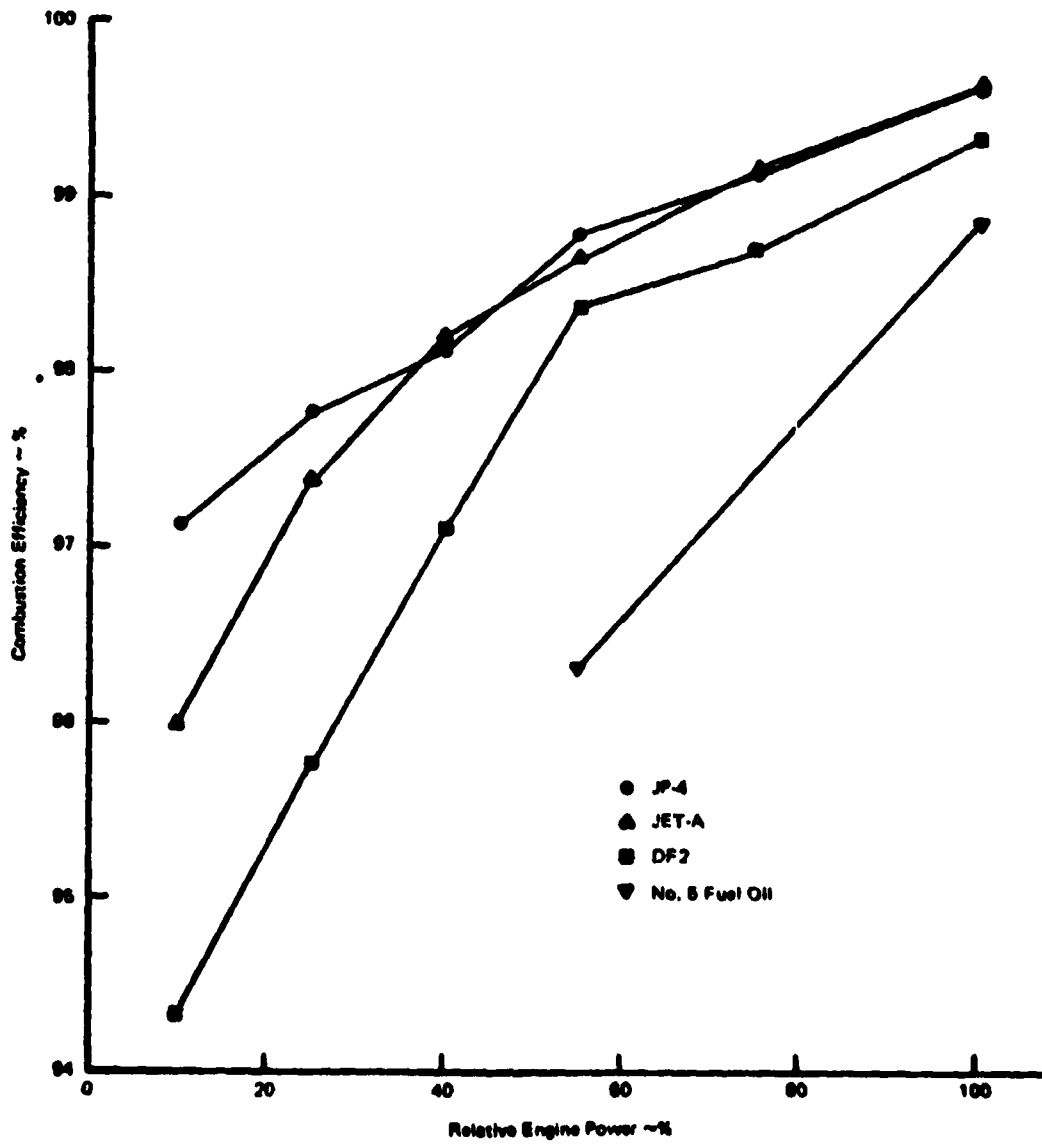
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Figure 4-2. Effect of fuel type on combustion efficiency for the
T-63 (Moses, 1975).

found to adequately represent the JP-4 data of Moses (1975) for the T-63. However, predictions for Jet A, DF-2 and No. 5 oil obtained from the mixing-controlled correlation were not satisfactory. The mixing-controlled model has been extended to include the heterogeneous effects observed in the T-63 data, but it is not as yet clear which of two models is physically correct. A separate effort is in progress to provide an experimental basis upon which to resolve this issue.

B. Mixing-Controlled Efficiency

A detailed investigation of species concentration and temperature fields within a disc-stabilized model combustor propane flame (Tuttle et al., 1977) revealed that exhaust plane emissions of carbon monoxide (CO) and unburned hydrocarbons (HC) are chiefly determined by mixing-controlled quenching of energy-releasing reactions in the shear layer adjoining the flame stabilizing recirculation zone. As cold inlet air is mixed with hot reaction products in this region, rates of reaction progress become slow in relation to residence times within the combustor, and unburned CO and HC are convected to the exhaust. Tuttle et al. (1976) modeled exhaust CO emissions by considering the kinetic rate obtaining in the shear layer:

$$\frac{d[CO]}{dt} \sim -f(T)[CO] = -[CO]/\tau_{co} \tag{4-1}$$

where the characteristic kinetic time for CO burnout, τ_{co}, is estimated as an inverse Arrhenius rate coefficient

$$\tau_{co} \sim \exp[E/RT] \ . \tag{4-2}$$

On integration from time zero to the mixing time available for CO consumption in the shear layer, $\tau_{s\ell,co}$, the expression becomes:

$$[CO]/[CO]_0 \sim \exp[-\tau_{s\ell,co}/\tau_{co}] \; . \tag{4-3}$$

Finally, expanding the exponential expression as a power-series in $(\tau_{co}/\tau_{s\ell,co})$, neglecting second and higher-order terms, and assuming that the initial concentration is dependent proportionally on fuel flow, the CO emissions index is given by

$$COEI \sim \tau_{co}/\tau_{s\ell,co} \; . \tag{4-4}$$

Thus, CO emissions increase as the required kinetic time and inversely as the available mixing time before quenching, as confirmed by Tuttle et al. (1976) for disc-stabilized flames.

Mellor (1977a, b) and Hammond (1977) extended this model to practical gas turbine can combustors. By considering a number of geometric variants of the T-63 combustor, Mellor (1977b) found that the mixing time could be estimated by the ratio of a characteristic length, ℓ_{co}, and the reference velocity

$$\tau_{s\ell,co} = \ell_{co}/V_{ref} \; . \tag{4-5}$$

ℓ_{co} was found to depend on combustor diameter and on an axial distance, ℓ_q, from the fuel nozzle to a site of massive air addition where quenching occurs:

$$\ell_{co}^{-1} = (d_{comb}^{-1} + \ell_q^{-1}) \; . \tag{4-6}$$

He suggested that although ℓ_q was generally the distance to the primary

holes, it could shift downstream at high power levels. Mellor and Washam (1979) in modeling the annular Pratt and Whitney JT9-D combustor also observed this behavior. They suggested quenching occurred at the location where overall equivalence ratio dropped below $\phi = 0.2$. Using this model and considering all the foregoing results they recommend the CO-emissions model found in Table 4-1. In the present study it has been found that this model also correlates AGT-1500 CO emissions.

Contours of CO and HC concentration within model combustors show a high degree of similarity (Tuttle et al., 1977). Mellor (1977b) indicates that exhaust emissions of CO and HC from practical hardware are algebraically related: combustion efficiency could be estimated in this way. However, Schmidt and Mellor (1979) have proposed a separate characteristic time model for combustion efficiency which also takes advantage of the observed relation between CO and HC. By analogy with the model for CO emissions the inefficiency is given by

$$(1-\eta_c) \sim \tau_\eta/\tau_{s\ell,co} \; . \tag{4-7}$$

Here τ_η is a kinetic time corresponding to that for CO

$$\tau_\eta = 0.01 \, \exp(4500(cal/mole)/RT_\eta) \, , \; msec \tag{4-8}$$

where the kinetic temperature is dominated by inlet air temperature

$$T_\eta = 0.9 \, T_{in} + 0.1 \, T_{exh} \; . \tag{4-9}$$

The mixing time is identical to that used in the CO emissions model. Schmidt and Mellor (1979) demonstrated this model for data from a simple disc-stabilized flame when evaporation was rapid in relation to kinetic

Table 4-1

Characteristic Time Models for
Mixing-Controlled Combustion

A. **CO Emissions Model**

$COEI$ (g CO/kg fuel) $= 35(\tau_{co}/\tau_{s\ell,co})$

$\tau_{co} = 10^{-3} \exp[10760 \; (cal/mole)/RT_{avg}]$ (msec)

$T_{avg} = \frac{1}{2}(T_{in} + T_{exh})$ (K)

$\tau_{s\ell,co} = \ell_{co}/V_{ref}$ (msec)

$\ell_{co} = [\ell_q^{-1} + d_{comb}^{-1}]^{-1}$ (cm)

B. **Inefficiency**

$(1 - \eta_c) \sim \tau_\eta/\tau_{s\ell,co}$ (percent)

$\tau_\eta = 10^{-2} \exp[4500 \; (cal/mole)RT_\eta]$ (msec)

$T_\eta = 0.9 \; T_{in} + 0.1 \; T_{exh}$ (K)

and mixing processes.

Using this model and kinetic time formulation the AGT-1500 engine test data of Fig. 4-1 (Marchionna, 1978) have been correlated by

$$(1 - \eta_c) = 2.73 \, \tau_\eta/\tau_{s\ell,co} - 0.02; \quad r = 0.92 \, . \qquad (4\text{-}10)$$

This is illustrated in Fig. 4-3. For these data the quench length was taken as the primary jet length ($\ell_q = \ell_{pri}$) for $\phi < 0.24$. For conditions above idle, $\phi > 0.24$ and the quench length was assumed to be the overall combustor length. (Thus the values of ℓ_{co} cited in Table 2-4 are representative of $\ell_q = \ell_{pri}$, and for $\phi > .24$, $\ell_{co} = 9.39$ cm corresponding to $\ell_q = \ell_{comb}$.) Note that for the correlated data any fuel type influences are adequately modeled through the flame temperature dependence of τ_η, and droplet evaporation processes are not limiting to efficiency.

The comparison between the data and the correlation are re-examined in Fig. 4-4. Here the efficiency data for JP-4 and DF-2 are plotted against the θ-parameter of Lefebvre (1966).

$$\theta = P^{1.75} \, A_{ref} \, D^{.75}_{ref} \, \exp(T_{in}/300)/\dot{m}_a \, . \qquad (4\text{-}11)$$

The mixing-controlled correlation provides excellent representation of the AGT-1500 data of Marchionna (1978). Several earlier AGT-1500 data for DF-2 (Marchionna and Opdyke, 1976) at lower θ values are also compared. The departure from mixing-control is illustrated by the discrepancy between the data and the model at $\theta = 0.02$, which is representative of the AGT-1500 starting transient. Here heterogeneous processes become important.

Also illustrated in Fig. 4-4 is a comparison of the correlation
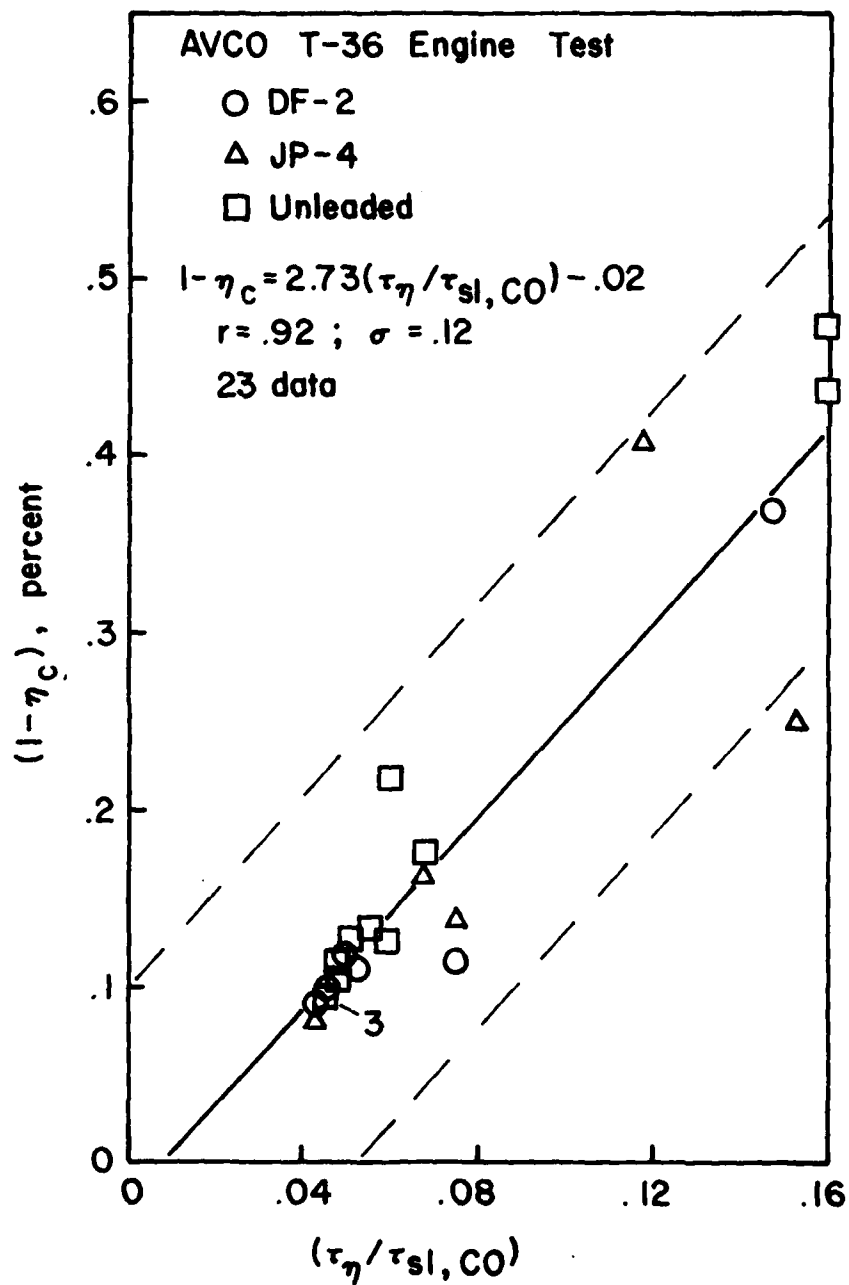
Figure 4-3. Correlation of AVCO engine test efficiency
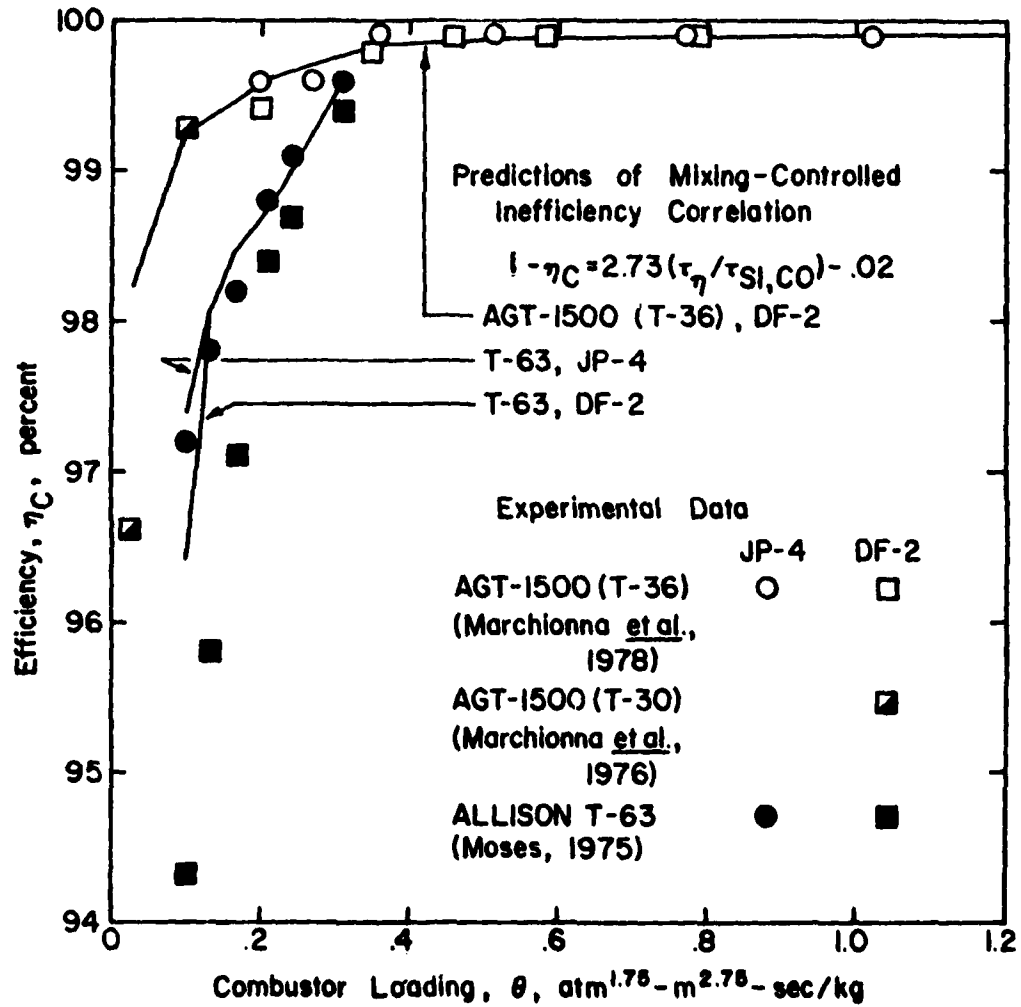data of three fuels for mixing controlled
combustion.

Figure 4-4. Comparison of present mixing-controlled efficiency data with experimental data for two fuels.

Eq. (4-10) and the JP-4 and DF-2 data of Moses (1975) for the T-63. The comparison for JP-4 is excellent over the entire power range. Although the mixing-controlled model predicts DF-2 efficiency will be lower at low θ-values (due to lower flame temperatures), it does not adequately represent the observed DF-2 efficiency data. At the low power conditions, the mixing-controlled model is in error by 2% efficiency (about 50% relative). Hence, although the AGT-1500 is not influenced by fuel vaporization over most of its power spectrum, heterogeneous processes become important at the starting condition. The T-63 engine is noticeably affected by fuel type selection, and the mixing-controlled efficiency model does not adequately represent these heterogeneous effects.

C. Heterogeneous Effects

In the absence of complete knowledge of how fuel properties and spray droplet sizes influence combustion efficiency, several models have been proposed to describe observed efficiency degradations. Each model incorporates the characteristic evaporation time, τ_{eb}, developed in Section II, to model heterogeneous effects. (For the present application the assumptions required to evaluate τ_{eb} are the same as those made previously to model lean blowoff.) Two of the models which have been evaluated adequately correlate the combined AGT-1500 and T-63 data presented in Figs. 4-1 and 4-2. A third model which has received wide dissemination (Ballal and Lefebvre, 1979c) is refuted here, based upon these same experimental data.

The latter model proposes that the combustion efficiency may be represented by the product of an evaporation efficiency, η_{evap}, and a mixing-controlled efficiency, η_{mix}, which may be considered to be a

function of the θ-parameter.

$$\eta_c = \eta_{evap} \cdot \eta_{mix} = \eta_{evap} \cdot f(\theta) . \tag{4-12}$$

The evaporation efficiency is defined by the ratio of fuel evaporated to fuel injected (Ballal and Lefebvre, 1979c), and, for a particular combustor may be written in terms of characteristic times (Leonard, 1980)

$$\eta_{evap} \sim \frac{\tau_{s\ell,co}}{\tau_{eb}} . \tag{4-13}$$

Hence, for a particular combustor and operating condition the combustion efficiency of one fuel relative to a second is given by

$$\frac{\eta_{c,1}}{\eta_{c,2}} = \frac{\tau_{eb,2}}{\tau_{eb,1}} \tag{4-14}$$

which, given the data of Moses (1975), is sufficient to evaluate this model, as shown in Table 4-2. Note that Eq. (4-14) predicts much stronger fuel type influences than are observed. This remains true if alternative assumptions are chosen to favor this model in evaluating τ_{eb}, including use of the stoichiometric flame temperature as the ambient evaporation condition, and complete neglect of drop size and Nusselt number differences. Indeed even the computations carried out by Ballal and Lefebvre (1979c) suggest DF-2 should only be about 75% as efficient as either kerosene fuel. Consequently the heterogeneous effects model of Ballal and Lefebvre (1979c) is rejected.

Two more acceptable models are based upon the mixing-controlled characteristic time model given above. Schmidt and Mellor (1979) suggested that the presence of droplets delays burnout of CO and HC, in

Table 4-2

Comparison of the Heterogeneous Effects
Model of Ballal and Lefebvre (1979c) with
T-63 Combustor Data*

Relative Power (%)	JP-4		Jet A		DF-2	
	η_c (%)	τ_{eb} (msec)	$\dfrac{\eta_c, \text{ Jet A}}{\eta_c, \text{ JP-4}}$	Eq. (4-14)	$\dfrac{\eta_c, \text{ DF-2}}{\eta_c, \text{ JP-4}}$	Eq. (4-14)
10	97.2	4.36	0.99	.63	0.97	.44
25	97.8	4.19	0.99	.63	0.98	.44
40	98.2	4.07	1.00	.63	0.99	.44
55	98.8	3.97	1.00	.63	1.00	.44
75	99.1	3.90	1.00	.63	1.00	.43
100	99.6	3.80	1.00	.63	1.00	.43

*Moses (1975)

effect increasing the kinetic time. This consecutive process model is expressed

$$(1 - n_c) \sim (\tau_n + k_1 \tau_{eb})/\tau_{s\ell} \tag{4-15}$$

where k_1 is a constant introduced to scale τ_n and τ_{eb} because the characteristic times are order of magnitude rather than absolute estimates. A second model suggests droplets cause additional deposition of CO and HC in regions where they are susceptible to quenching, in a process separate from and concurrent with the still important mixing. This separable process model is expressed

$$(1 - n_c) \sim (1 + k_2 \tau_{eb}/\tau_{s\ell,co})(\tau_n/\tau_{s\ell,co}) . \tag{4-16}$$

Here, the heterogeneous effect represented by τ_{eb} is referred to available shear layer mixing time since long mixing times in hot regions of the flame mitigate slow evaporation, hence limiting incremental deposition of the energy releasing species.

The consecutive process and separable process model correlations of the combined AGT-1500 (Marchionna, 1978) and T-63 (Moses, 1975) data are illustrated in Figs. 4-5 a. and b., respectively. Although the latter correlation achieves a slightly better correlation coefficient, this may not be significant statistically. For this reason correlation quality is considered in a more qualitative fashion in Figs. 4-6. Here each model is compared with corresponding T-63 experimental data by again plotting efficiency against the θ-parameter. The consecutive process model (Fig. 4-6 a.) is observed to reflect the fuel type influence reasonably well at high power levels and for JP-4 and Jet A even at low power levels. Considering the DF-2 data and correlation it appears, however, that the

$$1-\eta_C = 3.28\,(\tau_\eta + .03\,\tau_{eb})/\tau_{sl} - .17$$

$$r = .957 \qquad \sigma_y = .288$$

53 data

$I-\eta_C$, percent

$(\tau_\eta + .03\,\tau_{eb})/\tau_{sl}$

(A.)

$$1-\eta_C = 2.15\,(\tau_\eta/\tau_{sl})(1 + .12\,\tau_{eb}/\tau_{sl})$$

$$r = .979 \qquad \sigma_y = .098$$

53 data

$I-\eta_C$, percent

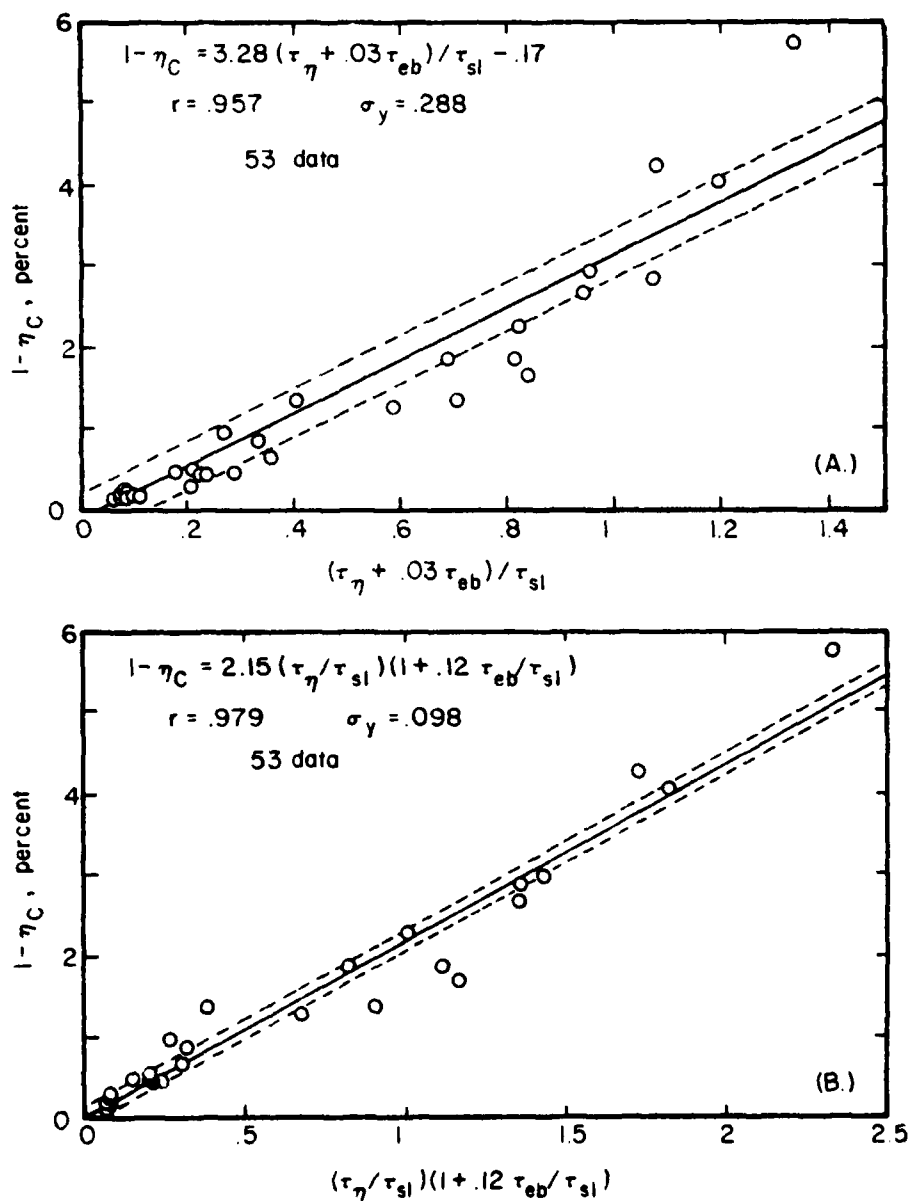$(\tau_\eta/\tau_{sl})(1 + .12\,\tau_{eb}/\tau_{sl})$

(B.)

Figure 4-5. Inefficiency correlations including heterogeneous effects of AGT-1500 (Marchionna et al., 1978) and T-63 (Moses, 1975) data by A.) consecutive process and B.) separable process model.
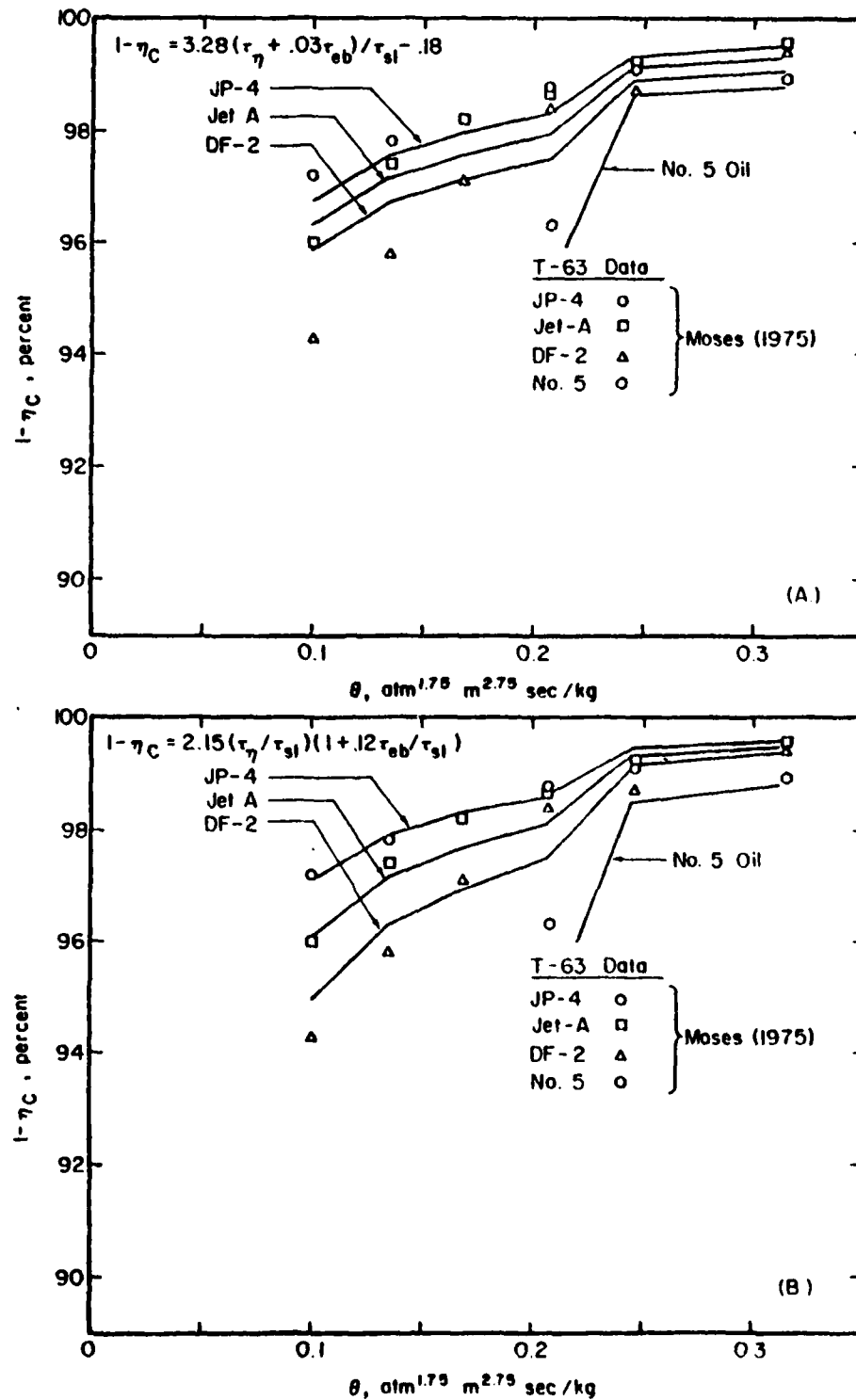
Figure 4-6. Comparison of combustion efficiency models with the data of Moses (1975): A.) consecutive process model and B.) separable process model.

character of the efficiency degradation is not well-embodied by the correlation at the lower power levels. In this regard the separable process model (Fig. 4-6b) appears to give qualitatively superior results at low power levels, but somewhat poorer agreement at intermediate power ($\theta = 0.207$ and $\theta = .246$). Some of this discrepancy may be due to the quench length shift (to the secondary holes) between these conditions being modeled as a discreet jump rather than a gradual shift.

In summary, the modeling of efficiency has progressed to the point where heterogeneous effects can be correlated in gas turbine combustors by either of two models. At present there is insufficient experimental justification to distinguish between them. Consequently, an experimental program is currently underway to provide the insight necessary to resolve this issue. Both input/output experiments, of the type conducted by Moses (1975), and detailed measurements of species concentration and temperature fields within representative disc-stabilized liquid-fueled flames are included. It is expected that comparisons of flame-structure data, obtained for several flames in experiments of the latter type, will reveal a physical justification for selection between these models, which can then be supported by correlation of the input/output data.

V. CONCLUSIONS AND FUTURE RESEARCH NEEDS

The lean blowoff and ignition models are quite similar in form and have therefore been combined in this discussion. The models describe the limits of lean blowoff and ignition through the competition of a mixing time and the sum of the kinetic and droplet evaporation times. Both models were developed with the aid of data from experimental rigs and have been extended to two can-type combustors, the Detroit Diesel Allison T-63 and the Avco-Lycoming AGT-1500, with a variety of fuels and geometric configurations. The significance of the models is that a designer can perform simple calculations to estimate the effect of fuel or combustor changes on lean blowoff or ignition without resorting to complex computer modeling via finite difference techniques or expensive experimental programs.

The accuracy of the models is primarily limited by inadequate knowledge of the drop sizes in a combustor. Both lean blowoff and ignition limits are quite sensitive to drop size, so one needs accurate estimates of fuel spray quality to properly predict the limits. These estimates are usually made via empirical equations developed for the nozzles in question. However, actual drop size measurements in combustors would definitely aid the lean blowoff and ignition work.

An additional area in need of further investigation is suggested by the similarities of lean blowoff and ignition. Note that lean blowoff curves are generally similar in shape to ignition limit curves and that stability limits occur at a leaner equivalence ratio than ignition limits (see Fig. 3-1). The similarity in shape of the curves should be expected since Eq. 3-1 was the basis for both the lean blowoff and ignition models.

This equation also indicates that the lean limit equivalence ratio should be less than the equivalence ratio required for ignition because the length scale for the mixing time is longer for lean blowoff (ℓ_{co}) than for ignition (d_q). Therefore, the ignition and lean blowoff models need to be examined simultaneously and the small differences between them eliminated to determine if a single unified model can be developed for both ignition and lean blowoff.

AGT-1500 combustion inefficiency at idle and higher power settings can be explained by mixing-controlled quenching of the final burnout of CO and HC. Heterogeneous effects associated with higher viscosity, higher boiling point temperature fuels are unimportant, and inefficiency is proportional to the ratio of a kinetic time and a mixing time characteristic of the quenching shear layer. This model is also applicable to other combustors, such as the T-63, when heterogeneous effects are unimportant. However, the T-63 is fuel sensitive, especially at low power. These heterogeneous effects can be correlated by either of two models which extend the basic mixing-controlled model using a characteristic evaporation time. Experiments are presently under way which will establish the physical mechanism by which heterogeneous effects become important and which will provide a basis for distinguishing between the heterogeneous effects correlations.

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